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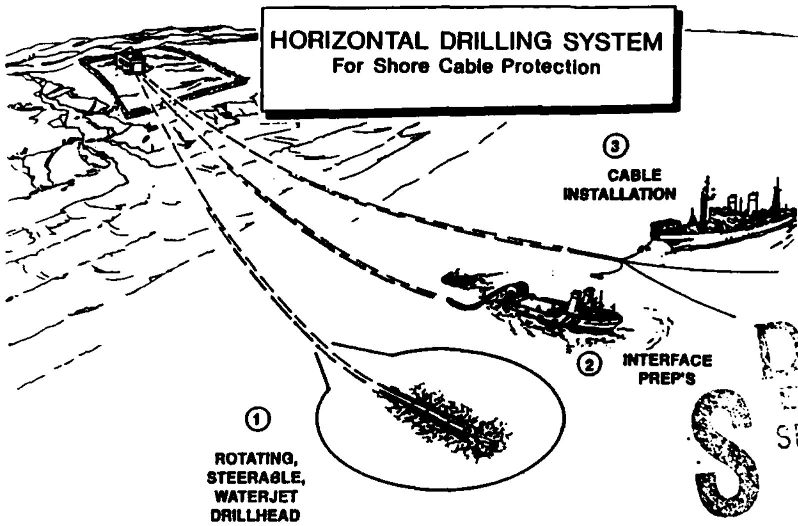
NCEL
Technical Note

HORIZONTAL DRILLING SYSTEM (HDS) OPERATIONS THEORY REPORT



ABSTRACT The Naval Civil Engineering Laboratory (NCEL) has developed a prototype horizontal drilling system (HDS) that is designed to reach distances of 10,000 to 15,000 feet and should be capable of reaching 25,000 feet in favorable conditions. This system uses high pressure water (15,000 psi) to "drill" at rates of 10 to 30 in./min. Steering is accomplished by instructing the drillhead (nozzle) to cut in an eccentric pattern. As part of a complete cable protection system, NCEL has also developed a technique for divers to adapt the seaward end of the drill pipe to a section of flexible pipe and a procedure for pigging multiple cables through the installed pipe assembly from shore or from sea. This allows easy interface between the installed shore cable section and the large cable ships that are used for deep water cable installation and deployment. Laboratory and limited field tests of the HDS system have been conducted and results agree with theory for the distance tested (500 feet). Negotiations are in progress to transfer the technology to private industry and continue the development.

NAVAL CIVIL ENGINEERING LABORATORY PORT HUENEME CALIFORNIA 93043-4328

METRIC CONVERSION FACTORS

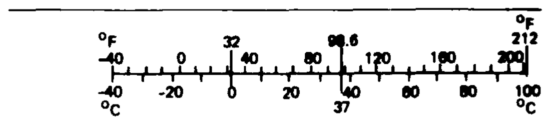
Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2,000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.

Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1,000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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The Naval Civil Engineering Laboratory (NCEL) has developed a prototype horizontal drilling system (HDS) that is designed to reach distances of 10,000 to 15,000 feet and should be capable of reaching 25,000 feet in favorable conditions. This system uses high pressure water (15,000 psi) to "drill" at rates of 10 to 30 in./min. Steering is accomplished by instructing the drillhead (nozzle) to cut in an eccentric pattern. As part of a complete cable protection system, NCEL has also developed a technique for divers to adapt the seaward end of the drill pipe to a section of flexible pipe and a procedure for pigging multiple cables through the installed pipe assembly from shore or from sea. This allows easy interface between the installed shore cable section and the large cable ships that are used for deep water cable installation and deployment. Laboratory and limited field tests of the HDS system have been conducted and results agree with theory for the distance tested (500 feet). Negotiations are in progress to transfer the technology to private industry and continue the development.

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1.0 INTRODUCTION

The Horizontal Drilling System (HDS) program was implemented at the Naval Civil Engineering Laboratory (NCEL) to develop an improved means of protection for shore landing cables. In particular, HDS developed a new technical approach using horizontal drilling to install a protective conduit (drill pipe) from shore to points far enough offshore to avoid the surfzone and reach water depths safe for a cable ship to moor (>60 feet). In many applications, the drilling distance required to reach safe water depths is significantly greater than commercial technology will allow. The present commercial limit for horizontal drilling is 5,000 feet (the same as it was when the HDS program began back in 1985). The HDS design goal was 10,000 to 25,000 feet. A successful system at those lengths would provide:

1. Ability to avoid the environmentally sensitive beach area.

2. Capability to reach under shore-fast ice.

A prototype horizontal drilling system was built for the HDS and successfully tested in laboratory and very limited field conditions. Unfortunately, because of schedule and fiscal limitations, the field testing was stopped before a complete, long-hole (>5,000 feet) test could be conducted.

Negotiations are in progress to transfer the technology to private industry and continue the development.

1.1 Operations Theory Report Outline

This report begins with a review of the objectives of HDS technology. The objectives are to install a cable protection system, consisting of an essentially horizontal hole, to long lengths, with reasonable accuracy and leave a casing in place in order to protect shore landing cable(s).

There are a variety of generic limitations on horizontal drilling. These are outlined, and a summary of the state of the art of commercial horizontal and directional drilling is provided. The limitations discussed focus on pipe stresses, usually in the form of buckling or joint failures. These stresses are caused primarily by frictional loads (from pipe weight, soil collapsed weight, or steering forces) or drilling forces.

Next, the general concept of the HDS technical solution is presented. A reference description of the system components, how they operate, and what part they play in the system function is provided. The key features of the HDS are:

a. Water Jet Drilling (to minimize bit push forces and add "rabbit force" tension to overcome buckling).

b. Rotating Drill String (to reduce push force friction).

c. Steerable Drillhead (to keep steering forces low and meet accuracy requirements).

d. Drill String as Casing.

e. Improved Joint Design (for greater torque capacity).

f. Logging Tool, Launcher, etc. (to operate while rotating).

The concept outline is followed by an analysis of the drill pipe operating theory. The analysis includes a listing of major variables, theoretical interactions, and limitations (particularly joint limitations and buckling). This section serves as the basic reference for future system design modifications.

Details are provided the Appendix and the references.

1.2 Limitations on Present Knowledge

For the most part, the present operating theory has not been validated and is still a "theory." Although there is a sound base of knowledge for such basic phenomena as water jet nozzles, hydraulic flow, joint stress modeling, and pipe frictional loading, there is much to be learned about the interaction of the drillhead and drill pipe with the soil or rock. Effective dynamic coefficients of friction for long horizontal pipe lengths rotating in various geologic formations have not yet been tested. The data are derived from lengths of only a few feet.

Steering phenomena have been addressed only briefly. Three test holes 400 to 500 feet long have indicated a general "natural" tendency to steer up and left, but only under limited conditions. Only one actual steering event has been attempted. Although it did successfully move the drill string in the direction intended, there is no calibration curve that can be used to determine in advance what steering command to give to produce a desired deviation in a particular geologic formation.

The models of effective increased bearing forces and friction as a function of steering are based on simple beam equations, but there is only one empirical data point to support estimates of proper lengths and deviations (radii of curvature) to use in operating the models as predictive or post-processing tools.

In addition, there will likely be many new lessons to be learned about the most practical ways to implement some of the theoretically desirable combinations of pressure, speed of advance, flow rate, and nozzle combinations. Only when these experimental, empirical experiences can be factored into the models will the operating theory be fully validated and functional. For now, the HDS "operating theory" is best viewed as:

a. A guide for collecting data.

b. A help in setting priorities when operating decisions have to be made.

c. A series of warning guideposts to help operators understand the risks they may be taking in the process of moving the technology forward to an operational capability.

2.0 OBJECTIVES

2.1 HDS Objectives

The HDS objectives are discussed in detail in Reference 1. They are listed here for reference:

a. Demonstrate distance (10,000 to 25,000 feet).

b. Demonstrate steerability with accuracy (±25 feet vertical, ±1,320 feet horizontal at 25,000 feet).

c. Conduct cable installation test (up to six 0.5-inch cables).

d. Demonstrate wet interface (flex pipe installation).

e. Provide personnel training on equipment.

2.2 Operations Theory Objectives

This operations theory serves several purposes. For an operational system, the operations theory is first used in the planning stages. The theory assists in determining realistic goals for the drill string installation, selecting sites that maximize the chances of a successful installation, determining lengths of pipe required, and developing a specific route plan to meet the protection system objectives. Initial positioning of the equipment, choices of nozzle configurations, estimates of water requirements, and many other specific design and logistic choices depend on an analysis of the expected route and drilling conditions. The theory provides guidance regarding the requirements for drilling fluid, requirements for casing, choices of drill pipe material, etc. The theory may even be used to establish maximum values and settings for protection (safeguards) of the system (allowable pressure, feed forces, rotation rates, etc.) during operations.

Once the installation is planned, the Operations and Maintenance Manual (Ref 2) is used to set up and check out the equipment.

As the drilling starts, information received from the Information Control Center (Ref 2) is regularly checked against the plan to determine required corrections. Whenever there is any change from the original plan, the theory is used to predict future loading and develop alternative plans as required. In essence, the theory allows calculations of ultimate limits (distance, pressure, torque, feed force) and also provides ways to look at alternatives. The theory, in its present form, does not provide a "best method" optimized solution. However, it does provide some general guidelines that lead to the most favorable approach. Also, it provides a means to make at least comparative evaluations of different drilling approaches.

Finally, the theory provides a format for evaluation of success in a completed hole, troubleshooting of problems, and development of improved methods for future operations.

During the remaining research and development phases of HDS, the theory is also useful as a tool to plan tests, provide on-site guidance during the execution of the tests, and evaluate results when the tests are complete.

In summary, just as the Operations and Maintenance Manual (Ref 2) serves as the technician's equipment setup handbook, this Operations Theory Report is intended to serve as the senior engineer's planning and operations handbook. This Operations Theory Report is the general framework and, together with the Operations and Maintenance Manual and the Field Test Report (Ref 1), provides the distillation of present understanding of HDS technology.

3.0 GENERAL LIMITATIONS

This section provides a general description of the known or theorized limitations on horizontal drilling in general, and specifically HDS technology.

3.1 Environmental

There are a variety of site-specific conditions that control the feasibility and overall performance of horizontal drilling efforts. These include geological effects, site topography/bathymetry, weather, covertness requirements, logistics (particularly water access), ecological concerns for local flora and fauna, and local politics. At this stage of development in the HDS technology, the data base on these particular problems is very sparse. However, since the HDS program went through the exercise of planning for a candidate operational site as part of the program and went through a limited version of these studies as part of Field Test I, the topics are worthy of discussion. These discussions also serve as an important place-holder for the insertion of future knowledge as the program continues. In time, this section could well become the largest part of the operations theory.

3.1.1 Geology. The ultimate limitations on drilling distance are derived primarily from the local geology. These limitations occur in the following ways:

a. The material and bulk strength of the formation determines the required water pressure at the drillhead. The fraccing pressure (the pressure required to fracture the surrounding formation) determines how much pressure must be applied at the launcher to produce a given pressure differential across the drillhead nozzles. For a reasonably consolidated formation, the available maximum launcher pressure of 15,000 psi is ample to cut natural rock formations up to about a 30,000-psi material compressive strength (granite). Fortunately, most natural formations are already fractured well enough that the cutting effect is controlled by the bulk strength, which is generally much lower than the material compressive strength. Experience in Field Test I showed that even a reasonably consolidated formation of cherty to shaley limestone could be cut at pressures of 8,000 to 10,000 psi (Ref 1). Even though the material compressive strength was about 25,000 psi, the bulk strength was typically 3,000 to 10,000 psi. At the full system pressure of 15,000 psi, the drill could be advanced at near maximum launcher speed (about 36 in./min). Only when occasional solid rocks were encountered did the system need to slow down. Even then, with full pressure, the drilling rate could be maintained at 3 to 6 in./min. High bulk strength only becomes a problem when working at very long drilling distances. Under those conditions, it becomes possible to have a situation in which the geology requires a high cutting pressure and torque. This increases the risk of pipe joint failure. The pipe joint is the weakest link in the drill pipe. This problem is discussed further in Section 5.4.

b. Geology is also a problem if the formation is too soft. There are two problems related to soft, unconsolidated soils. The first is hole stability. If the hole collapses, the torque and push force loads on the pipe per foot increase dramatically. Forces become a function of the density, depth, and length of the collapsed soil rather than just the weight of the pipe. These collapsed-hole forces can easily be an order of magnitude greater than the simple friction loads in an open hole. This consideration suggests planning operations in well consolidated sediments. To maximize distance, it is better to select hard soil and drill slowly than to select soft soil and try to drill rapidly. If there are sections of soft material that cannot be bypassed, the options are to use casing (if it is near the start of the hole) or to use drill mud to help stabilize the hole through that section. The effects of collapsed holes are discussed in more detail in Section 5.3.4.

The second problem with soft soil is at the exit end of the pipe. In some cases, the exit area is the seafloor. If the soil is very soft, the pipe will not be able to support its own weight and climb out onto the open seafloor. Of course, a small amount of sediment or sand covering the end of the pipe can easily be cleared by divers. However, if the pipe is several feet below the seafloor, the excavation becomes prohibitive. The solution to this problem is to aim for an area of seafloor that is consolidated and exit up toward the seafloor at a reasonably steep angle. The angle is determined by the expected depth of soft material and the stiffness of the drill pipe (Section 5.1.3). Finally, it is important for divers to inspect the exit point WHILE THE PIPE IS STILL ROTATING (although with limited water flow, for safety). This way the pipe can be pushed further (or even pulled back) to a suitable working configuration before stopping the pipe and allowing the cuttings to consolidate and lock up the pipe in the hole.

3.1.2 Site Topography/Bathymetry. The HDS drilling system is not adapted to making sharp turns. Fortunately, gravity does aid pipe bending. It is therefore easier to make turns (without adding frictional loads) in the vertical plane than in the horizontal plane. Even at that, it requires up to 180 feet of pipe for a 2-degree climb to avoid additional torque or push-force loads. Further, the present HDS launcher and foundation assembly are designed primarily to develop horizontal loads and operate horizontally. Therefore, the present system is not configured to start at large downslope angles (γ). That means if the launcher elevation is much above the shoreline, a relatively large distance between the launcher and the shoreline is needed. For example, assuming the drill pipe must pass at least 40 feet below the beach, Table 1 shows the approximate required distance the launcher must be back from the beach for various elevations.

Note that in the 0 elevation case, the required large turning radius adds so much length that increasing the dip to 7 degrees actually increases the standoff distance for small launcher elevations.

In general, increasing distance from the beach is desirable because it decreases environmental impact, but it is undesirable because it increases costs (more drill pipe, longer drilling operations). If water is drawn from the ocean, standoff distance also increases the costs of this operation. Picking the best starting point requires careful analysis of many factors.

The ocean bathymetry has a similar effect on installation planning. A steep slope on the seafloor makes it possible to reach working depths with a shorter drill pipe (lower costs). However, it also increases the visibility of offshore operations and makes their connection with shore operations more detectable. The ideal seafloor slope is one that is flat out to a reasonable

Table 1
Required Distance of Launcher from Beach (Y) (ft) Versus
Elevation (Z) and Entry Dip Angle (γ)

Entry Dip Angle (γ)	Elevation of Launcher (Z) (ft)						
	0	20	40	60	90	160	210
2 degrees	1285	1858	2430	3003	4435	5867	7299
5 degrees	807	1035	1264	1493	2065	2636	3207
7 degrees	815	978	1141	1304	1711	2118	2526

distance to provide a covertness disconnect, but remains well within drilling distance limits (e.g., 10,000 to 15,000 feet) and then slopes 5 to 10 degrees down. This provides a clean exit point without a lot of steering, minimizes chances of an excess of soft soil at the exit, and still allows reasonable mooring conditions for the offshore work vessel(s). At the very least, the seafloor bathymetry at the exit point should be smooth enough over the target area to allow easy location of the drill pipe on the seafloor, access for diver operations, and a straight installation for the flexible pipe. Also, the seafloor exit point needs to have enough visibility for safe diver construction operations.

3.1.3 **Weather**. Since the launcher area is covered to provide physical security, weather is generally not a problem for the drilling operations.

Weather can be a problem if the temperatures are very cold or hot. The Field Test at Naval Weapons Center (NWC) China Lake, California, in the summer had problems with overheating of the hydraulic power units and some of the electronics in the steerable drillhead. These are being corrected, so heat should not be a limiting factor in future operations. Extreme cold is a problem primarily for the stored water. Other equipment is inside or generates heat while running, but the water storage will need to be kept above freezing. The choice between preheating or adding antifreeze will be site specific and based on economic and environmental considerations.

If there has been extended cold weather so that permafrost is created, there may be problems at two levels. The launcher site must be designed to ensure that there is no settlement when the permafrost is thawed. The drilling operations will have to be modified so that the heated water traveling down the drill pipe doesn't freeze up. Permafrost may provide very high fraccing pressures. It also may be a problem whenever flow rates are reduced (when stopped to add another pipe section, or for repairs). It may be necessary to preheat the low-flow maintenance water. This type of operation would be a good subject for some special testing and analysis before undertaking a full field trial in Arctic conditions.

The offshore weather is probably the most critical weather problem. The diving operations to confirm proper exit and the construction operations to add the flex pipe all require reasonable visibility for safety. They also require reasonable surface conditions for a moored

vessel to conduct diving operations and handle the flex pipe. Sea state 4 is probably the effective limit, with sea state 3 much preferred.

Visibility below 3 to 5 feet is unacceptable for the original inspection dives (rotating pipe). Visibility below 1 to 3 feet is unacceptable for the construction work.

3.1.4 Logistics. There are four primary logistics concerns:

a. Construction/location of the launcher site.
b. Delivery of pipe.
c. Delivery of water/fluids.
d. Housing/support of personnel.

The logistics problems for the construction of the onshore work site are the normal problems with transportation of heavy construction equipment. They are, of course, very site specific. In general, it is best to first erect a large shelter to cover the main launcher site or, better yet, work inside an existing building. However, it may be possible to install the main concrete pad first as was done in the Field Test. If the construction is conducted at a reasonable distance from the beach, on some existing Government facility, using normal construction equipment, the logistics problems are minimal.

Delivery of pipe can be complicated. The best approach is to have the covered pipe shipped to arrive in stages over as long a time as is practical. This reduces the space requirements for staging at the site. Pipe can be brought inside the launcher building covered and then handled in separate sections as needed.

The water requirements for a HDS can be a considerable problem. The hardware can operate with freshwater, seawater, or other fluids. However, the daily requirements are large. The maximum requirement is derived as follows:

a. Maximum flow rate = 200 gpm (12,000 gallons per hour).

b. Best-guess operating duty cycle, in slow-cutting soils, with no logging being done, working 24 hours per day is about 75 percent, or 18 hours per day.

c. Expected MAXIMUM water usage = <u>216,000</u> gallons per day.

A more normal usage expected during a field test, or if the system cuts faster and cycles more slowly than above, would be an operation at about 150 gpm (one pump) - one logging operation per day, at a duty cycle of about 50 percent, or about 10 hours per day. This scenario produces:

Expected NORMAL water usage = <u>90,000</u> gallons per day

Water storage requirements depend entirely on the available water supply. The facility could range from a small tank to serve as a backup of low pressure water in case of interruptions, to a major storage facility to carry several days' supply. At the Field Test at NWC China Lake, a well was drilled and there was also on-site storage because the well could meet normal requirements but not maximum requirements.

As with pipe, trucking in water or draining large amounts from a local supply could affect the environment. However, if there is a piped source (from a dedicated well or the sea), the effect of water input is minimal.

3.1.5 Ecological Concerns. A major strength of HDS technology is that it has virtually no visible impact on the beach or surfzone. However, during the drilling operations water or other fluids are injected in considerable quantity below the surface. For example, if the drilling averaged 150 gpm, at a drilling speed of 6 in./min, that would be 300 gallons of water for every foot drilled. Assuming the soil is as much as 10 percent void, that equates to a fully-saturated cylinder 22.5 feet in diameter all along the pipe length. Tests at NWC China Lake confirmed that water could reach the surface when the pipe was 10 to 30 feet below the surface. The possible impacts include damage to surface vegetation, flooding of animal burrows, slumping of soils, etc. The environmental impact is greatest where the water is other than freshwater. In desert areas, even the residual clay deposits left after water dries on the surface may be considered unacceptable. While it is generally true that HDS drilling is not doing anything to the environment that can't be matched (and erased) by a good rainstorm, the issues must be considered.

3.1.6 Local Politics. This is a catch-all topic, but it includes a variety of potentially significant issues. The HDS technology requires some land space on shore; Field Test I camp was about the size of a football field. The logistics of water and pipe supply may affect local traffic and residents. The drilling injects water and possibly other materials into the underground, possibly near aquifers, and the water can escape through fractures to the surface from depths of 30 feet or more. There is noise from the diesels, and the drillhead produces a sound that is detectable from 10 to 50 feet away, depending on the acoustic properties of the soil formation. There are nitrogen-oxygen and other air pollutants emitted from the diesels. If private lands lie between the launcher and the beach, there are issues of rights of access, infringements on mineral rights, etc. These issues can affect the required depth of the drill pipe, which can affect the entire drilling plan. The entire issue of operating permits depends on the ownership of the land, level of environmental impact, etc. The permitting process can have a major impact on schedules and costs.

The details of experience to date with these issues are contained in Reference 1, the HDS Test Report. The issues are raised here only to remind the reader that engineering analysis and drilling considerations alone are not sufficient to produce a viable plan for an HDS installation (or even a full-scale field test). The social aspects of the operation should be considered before the detailed engineering analysis is even begun.

3.2 Pipe Stresses

Assuming that all the other site-specific social, logistic, and technical problems have been overcome, the ultimate limitations on HDS technical performance appear in the form of pipe stresses. No matter how well the launcher, pumps, and instrumentation perform, there are physical limits on the distance and route that the HDS drill string can achieve. In fact, all the support systems are designed to recognize and minimize those pipe stresses, and to limit themselves before the stresses lead to pipe failure.

The primary failure modes caused by pipe stresses are:

1 = 0.100 inches (5-degree angle)

a. Joint Failure. This is caused when the applied torque causes the friction washer to slip and/or yield the pipe joint material. Joint rotation then rapidly applies very high compression to the box and tension to the pin and the pin fails. It also is possible to fail the joint through "mushrooming" the pipe faces against the washer. This is discussed in detail in Section 5.4 and Reference 1.

b. Pipe Buckling. The pipe may buckle as a free column (in the sections within the launcher or between the launcher and the hole) or sinusoidally within the hole. By careful design of the launcher supports and by minimizing hole diameter, it is possible to keep buckling limits high enough to allow reasonable speeds of advance. In any event, as long as the pipe is rotating, it is always possible to slow down the advance rate and reduce the push forces to an acceptably low value. The only limit on this is if the hole collapses over a long length. Then it would be possible to develop a situation in which either torque is too high for the joint, or the possible speed of advance is too slow to be economically feasible. This is discussed in detail in Section 5.5.2.7.

One fortunate aspect of pipe stresses is that they are theoretically always maximum at the launcher end of the system. Of course, there is the possibility of a material flaw, or of some rare local wear point causing a failure somewhere down hole. For example, in the test program, an adaptor end failed because of a weldment and design problem. However, for built-as-designed equipment, the stresses are maximum at the launcher end. The advantage to this fact is that IF there is a failure, the pipe is likely to be accessible for redressing, reconnection, and restart of the operation at a lower force level (or recovery of the string).

Also, it is generally true that it is not likely for the system to push itself into a situation from which it cannot be extracted. The pipe tolerates much more load in tension than in compression. Since a withdrawal of the string always occurs at low water pressure, the joint always has more residual strength in a pull-back mode than in a drilling mode. Also, the launcher itself is designed to provide greater pulling force than pushing force.

Further, experience to date shows that if the system is being overloaded by excessive steering, too much advance rate or other such commands, the unit can always be slowed down and it will relieve itself. In general, a hole that is too curved is "straightenable" by a slow, high pressure pass. In fact, about the only tight spot envisioned so far is a case in which the string encounters an unexpected very hard formation, near the end of the run (just before seafloor exit). If the joints will not tolerate enough pressure to cut through that spot, the string has to be pulled well back and another exit location attempted. The steering mechanism of the steerable drillhead requires a reaction force against a side wall to deviate the pipe. In a predrilled hole, the only reaction force available is gravity. With enough time and cutting, the theory is that the pipe would eventually sag enough to develop a new downward hole. From that hole, the unit could be steered left or right to miss the obstacle. This concept has not been tested.

3.3 Conventional Limitations

There is a major specialty group within the drilling industry that specializes in horizontal drilling. There are three major subsets of this technology:

a. Directional Drilling. In this technology, the hole starts vertically and then deviates from the vertical. Usually, the deviation is less than 45 degrees from the vertical, but the

technology is also used to reach all the way to horizontal. In this approach, the end of the drilled hole (bottom of the well) is always many feet below the surface.

b. Slant (or Inclined) Drilling. This is the same as for directional drilling except that a special drill rig is used to start the initial hole as much as 30 degrees from the vertical.

c. Horizontal Drilling. This technology uses a rig similar to that used in slant drilling but starts nearly horizontal and ends up with the hole on the surface.

All of these systems use mature technology, and several commercial sources are available. The Naval Facilities Engineering Command (NAVFAC) has recently used a commercial source (Cherrington) to install a 3,000-foot, 4-inch pipe sewer outfall at Centerville Beach, California. Unfortunately, commercial technology today cannot reach the horizontal distances required for most Navy or other cable shore landings. Figure 1 shows the limitations of these technologies. Figure 2 shows more details of the various types of horizontal drilling. In general, the maximum horizontal distance that is commercially available for a surface-exiting hole is 5,000 feet. There have been cases cited of a 7,500-foot pilot hole for a Japanese tunnel and a 10,000-foot installation in Holland. However, these occurred under very favorable geological conditions and are considered exceptions. In any event, HDS goals start where even these exceptions leave off, i.e., 10,000 to 25,000 feet.

The basic limitations on commercial technology are all related to pipe buckling. None of the horizontal systems require continous pipe rotation to reduce the coefficient of friction. However, most of them require considerable pushing force at the bit to provide cutting. The few that use water jet cutting are limited both by the buckling problem and the allowable stresses in the pipe joints (high pressure and compression).

The HDS attempts to use the experience gained in the commercial world to provide a new combination of technologies to significantly extend the state of the art.

4.0 HDS SOLUTION CONCEPT

Figure 3 is an overview of the HDS application. HDS provides protection to one or more shore cables by installing them in a long drill pipe under the nearshore surfzone and other threats.

The HDS is capable of horizontal distances several times greater than commercial horizontal drilling systems due to the following:

a. A continuously rotating, steerable water-jet drilling system that leaves the drill string as the installed casing.

b. The high-torque joint and a high-accuracy logging system that functions while the drill string is rotating.

c. A wet-end interface system using flexible pipe. This system allows installation of the cables by pigging from shore, and connection of the cables by conventional cable ship methods.

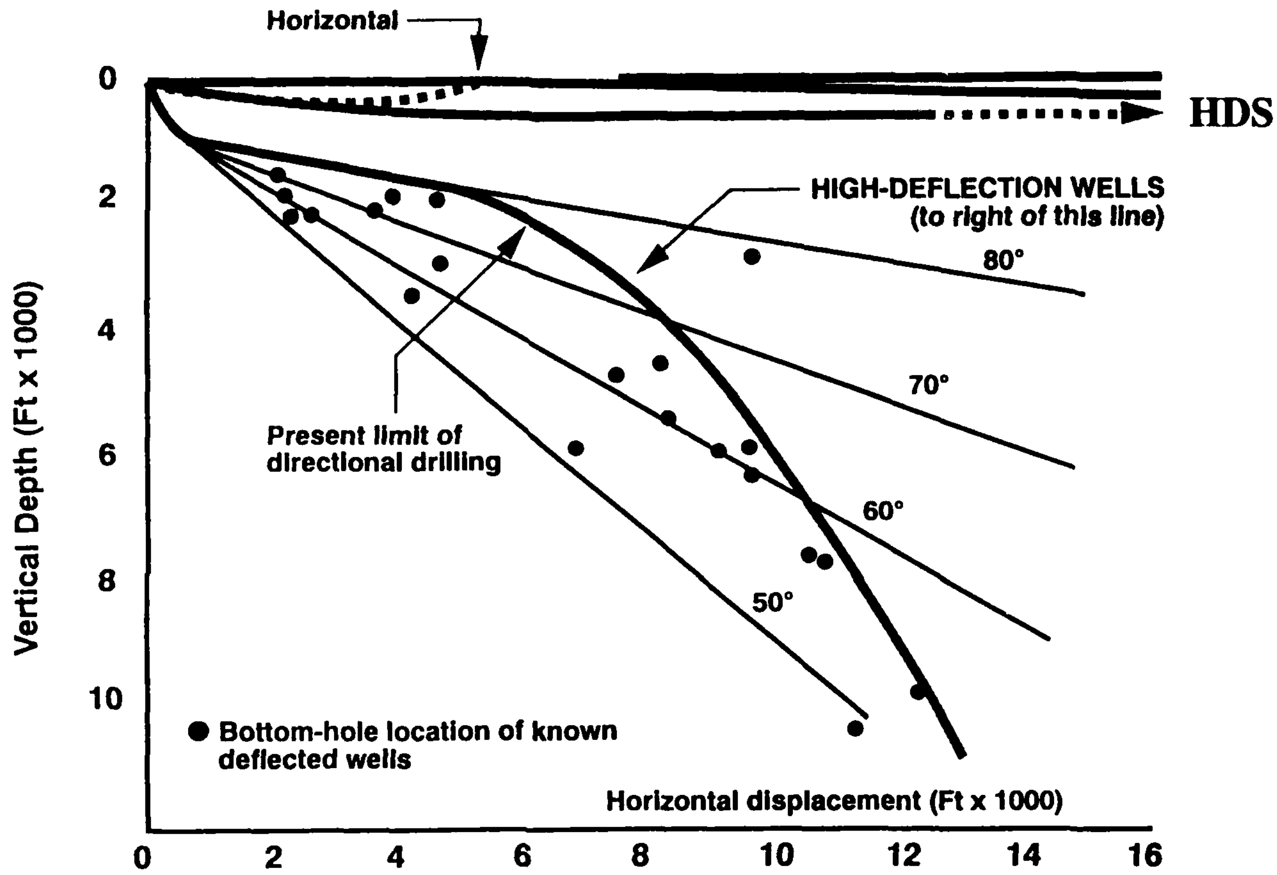


Figure 1
Comparison of drilling systems.

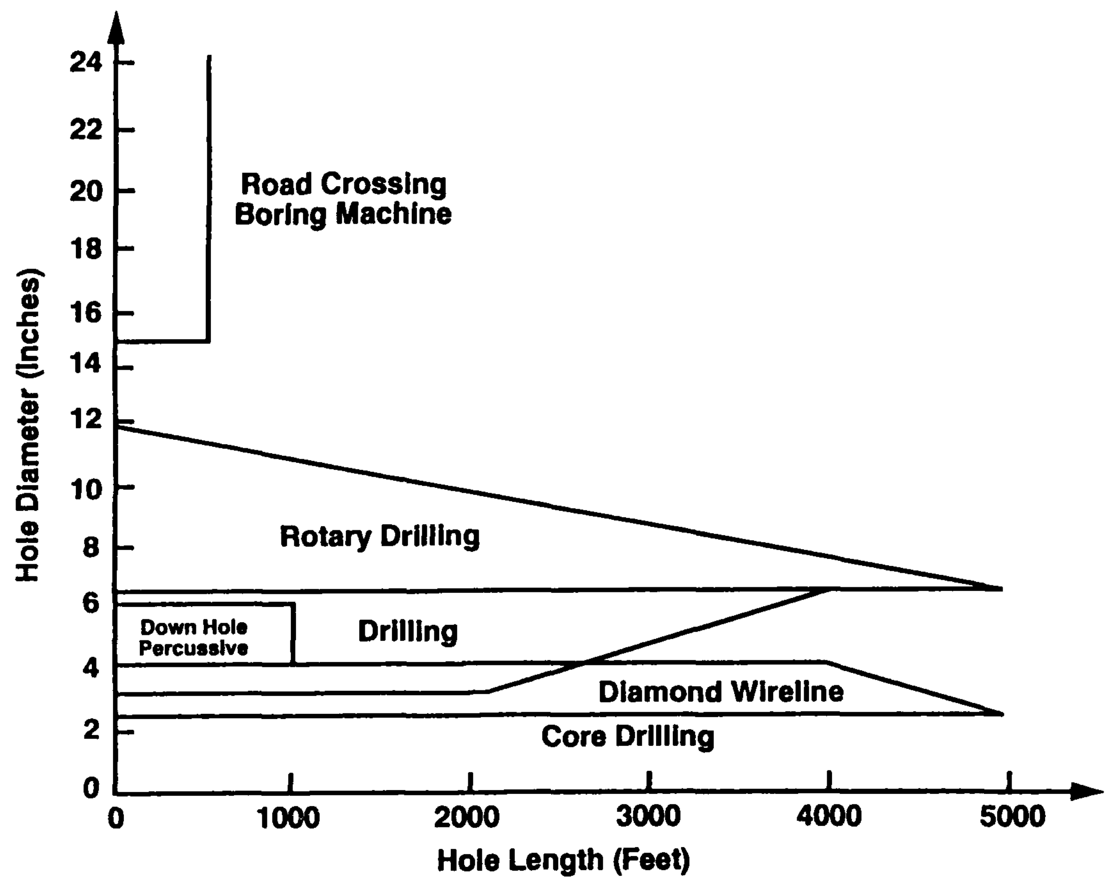


Figure 2
State-of-the-art horizontal penetration capability (Nov 1991).

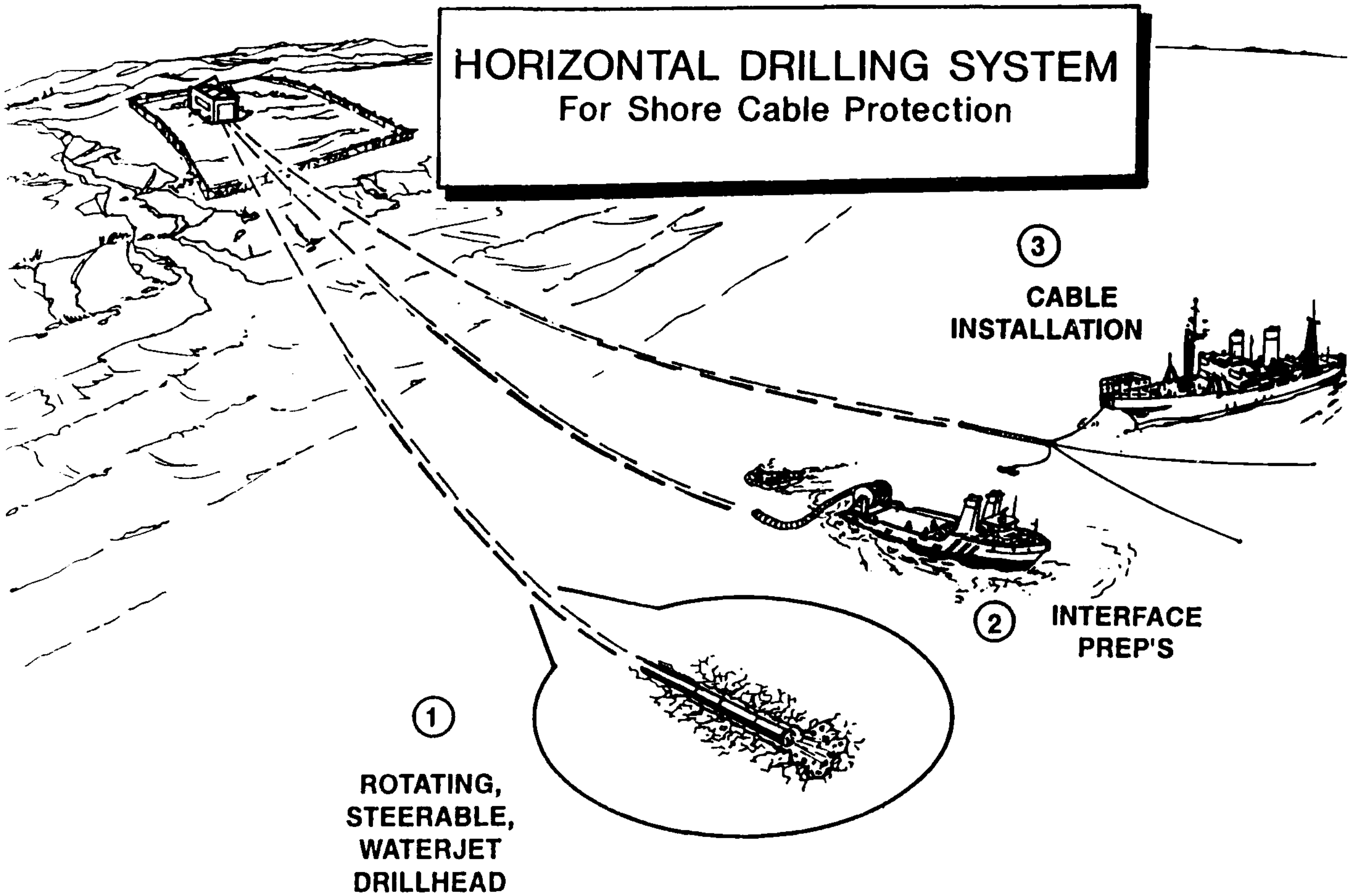


Figure 3
Horizontal driling system for shore cable protection.

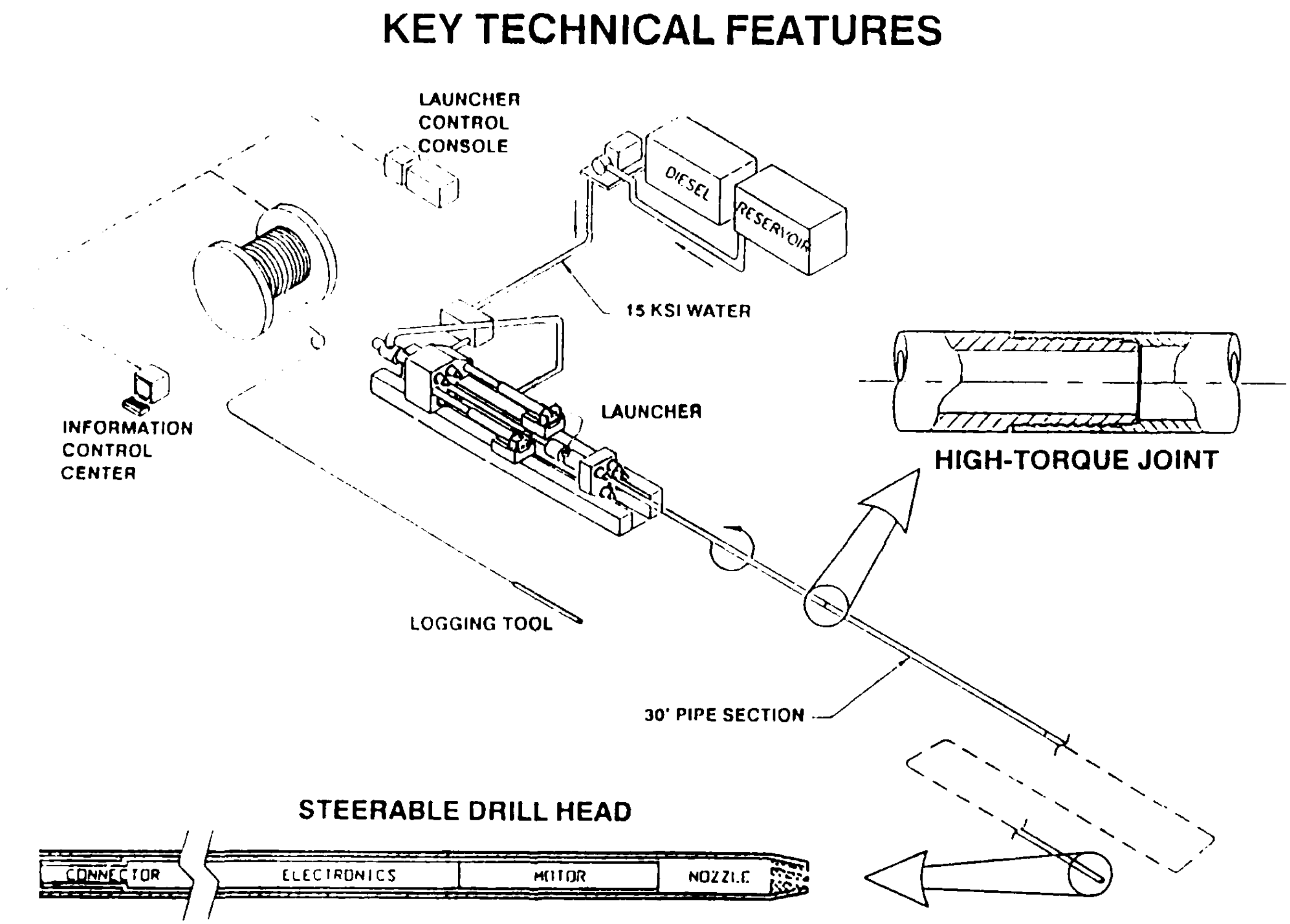


Figure 4
Horizontal drilling system key technical features.

The HDS is composed of the following equipment (refer to Figure 4):

LAUNCHER. The launcher is a hydraulic machine that provides rotation (0 to 9 rpm), torque (60,000 ft-lb), and push/pull forces (280,000/320,000 pounds) to the drill string during drilling operations.

DRILL PIPE. The drill pipe is a high strength drill pipe made from 4145 steel alloy with a 4.75-inch OD and a 3.5-inch ID. The drill pipe includes a specially designed flush joint with a designed torque capacity of 37,000 ft-lb at 15,000-psi internal pressure and zero push force.

STEERABLE DRILLHEAD. The steerable drillhead provides steering control to the drill string. This unit is battery operated and can be instructed to cut/steer in any one of the four quadrants - up, down, right, or left.

LOGGING SUBSYSTEM. The logging subsystem provides real-time location information for the drill string. The logging subsystem includes a logging tool, logging winch, logging cable, and data acquisition control center.

INFORMATION CONTROL CENTER. The information control center (ICC) records system parameters (flow, pressure, torque, and forces) during drilling operations. It also contains system control circuits to protect the drill string from failures due to exceeding the limits on one or a combination of various parameters: pressure, torque, and push forces.

DIESEL PUMP ASSEMBLY. A 1,000-horsepower diesel pump assembly provides up to 15,000-psi water pressure to the drillhead. A second diesel pump unit is provided as a backup unit.

BOOST PUMP AND FILTRATION SUBSYSTEM. The boost pump subsystem provides boost pressure to the high pressure pump assemblies. It also filters the water prior to it reaching the high pressure pump.

The remainder of Section 4 outlines the functions of the HDS hardware and software components. It includes the existing knowledge about selection of components to optimize system performance. Section 5.0 then addresses limitations on system performance, especially those related to ultimate load capabilities of the drill string itself.

4.1 Water Jet Cutting

Figure 5 shows the general arrangement of the drillhead assembly at the end of the pipe. High pressure water jets are used to cut the rock or soil. As the drill string rotates, each jet cuts a circular part of the end of the hole. The cuttings are maintained in a slurry and are forced back along the drill string. Some of them remain in the hole, but most of the fine material is carried out into fractures in the formation along with the drilling water. Note that this system requires an appropriate distance between the drillhead and the end of the hole to work. If the drillhead is too close, the angles of the nozzles do not allow the hole to be cut to a large enough diameter to pass the drill string and return the cuttings. If the drillhead is too far back, the hole

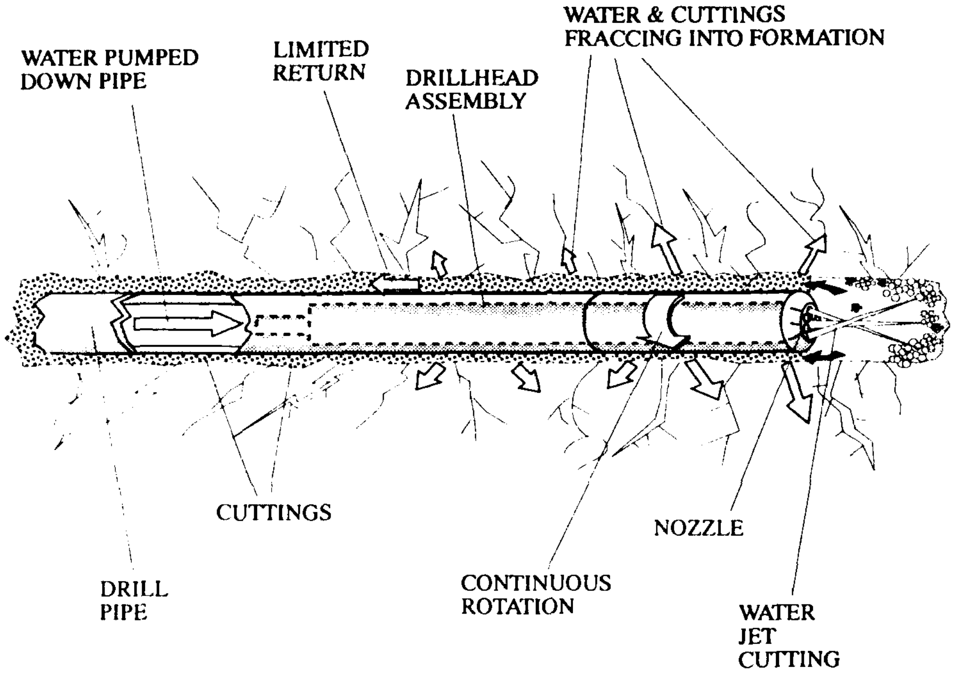
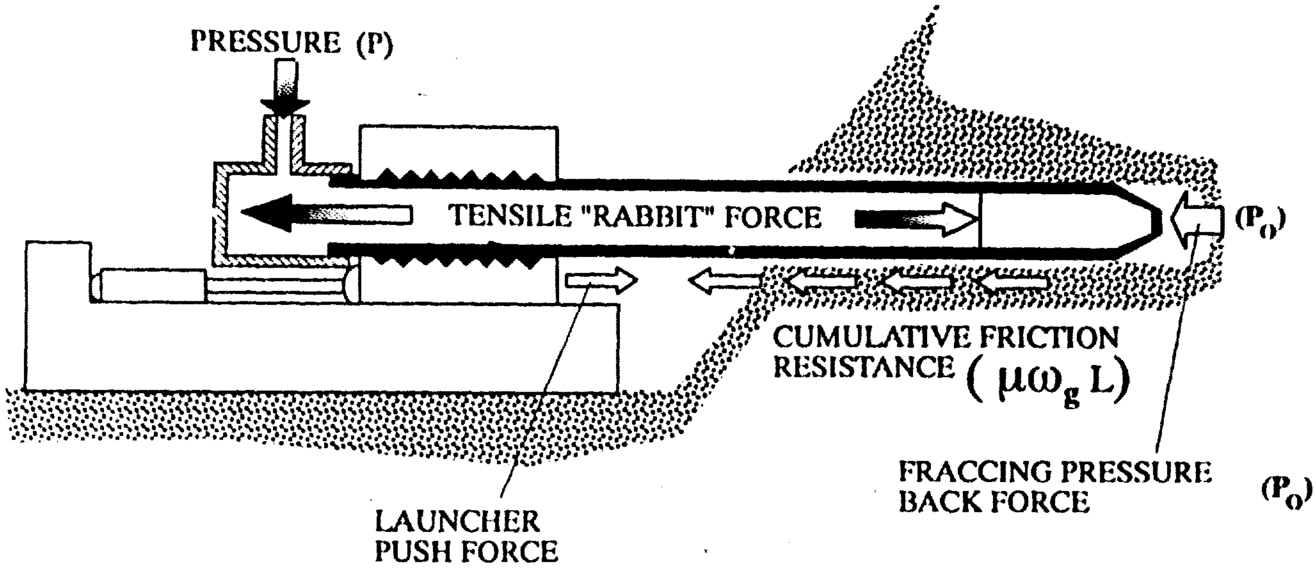


Figure 5
Water jet cutting drillhead concept.

enlarges and eventually all cutting stops because the jet velocities reduce below cutting levels. Balancing the drill string forward speed against the cutting speed resulting from the combined pressure and flow of water is the essence of water-jet drilling.

4.2 Pressure to Tension Pipe

The internal pressure that provides the velocity and flow to cut the soil or rock formation is also an important key in managing pipe stresses. Figure 6 shows that the internal tension provided by this pressure helps overcome the compressive forces required to move the pipe forward against friction. In general, as long as the nozzle end is not in contrast with the end of the hole and the drill string is not moved forward at high speeds, the tensile loads due to internal pressure will exceed the compressive loads and the pipe will not buckle. This is discussed in detail in Section 5.0.



$$\text{TENSION}\ (PA_i) > \text{COMPRESSION}(\mu\omega_g L + P_0 A_0)$$

Figure 6
Rabbit force concept.

4.3 Rotation to Reduce Friction

Even with internal pressure, it is not possible to reach extremely long drilling distances without one other important step. Section 5.0 develops the principles that show how maintaining continuous rotation of the pipe reduces the basic coefficient of friction and also allows very small push forces to maintain the pipe in forward motion. The price that is paid is the complexity of continuous rotation. The drillhead must be designed to steer while rotating, the logging system must function while the pipe is rotating, and joints must be made and broken while rotating. Analysis and early tests show that the pipe can be stopped for brief periods, particularly during the early stages of drilling, and restarted without difficulty. However, in the final stages of drilling over long distances (> a few thousand feet) continuous rotation will probably be required.

4.4 Steerable Drillhead

The drillhead is not just a single element. It is actually a series of hardware components that can be configured in many ways. The selection of the proper configuration for any particular drilling evolution is critical to success.

There are two standard types of drillheads used for the drilling operations:

a. Straight-ahead drillhead (SADH)

b. Steerable drillhead (SDH)

The straight-ahead drillhead is a simple tool, with no moving parts except the guide rollers used to guide it down the pipe. It is used for most of the drilling operations. Figure 7 is a photograph of the present SADH.

The SDH is a much more complex assembly. Figure 8 shows the drillhead terminology. In HDS terminology, the "drillhead" refers to the top-level assembly that is handled in and out of the drill pipe. It contains, among other things, the nozzle and its associated orifices. The "nozzle" is the solid block that contains the orifices. It seals against the inside of the adapter section at the end of the drill string. The leading portion of the nozzle is a 16-degree tapered surface, which seats against a mating surface at the end of the drill string, sealed by a bulk rubber seal.

Inside the nozzle are several "orifices." These are small metal inserts with various sized holes. They are used to set the actual hole size (and thereby, pressure and flow rate) through the nozzle. The nozzle has four hole positions to receive the orifice inserts. Each position may have orifices of various sizes in it, or it may be plugged entirely.

The SDH uses the same basic cutting orifices as the SADH, but they are backed by an electrically-driven motor assembly that is capable of opening and closing selected orifices as they pass selected orientations during pipe rotation. It is this selective cutting that provides the elliptical or off-center hole that creates steering. The SDH also contains electronics that can receive and store program commands for use in providing timed steering maneuvers. The SDH also can respond to commands sensed from the drill string rotation rate.

While the two drillhead configurations presented in detail below offer a wide range of diverse water-cutting capabilities, they are not without need of further investigative analyses. Experience to date is limited, so each time a new setting is selected it will need to be carefully monitored until a broad experience base is developed for this new drilling method. Different nozzle sizes and angles should be examined with corresponding changes of water pressures and flow rates in various drilling conditions.

Two areas of consideration for drillhead selection are:

a. Orifice Size. Orifice size selection is dependent upon the anticipated water cutting pressure and flow rate to be used.

b. Orifice Angle. The angle cut of the orifice has direct bearing on the type and size of hole desired. The more straightforward orifice makes a direct frontal cut; the angled orifice creates a wider cut. Testing has shown that harder material requires wider orifice angles.

Several different types of cutting orifice sizes and angles are employed to achieve the maximum drilling rate performance.

The equipment operator has the following measurable parameters to indicate the actual drilling conditions and performance during the drilling process:

- Water pressure
- Drill pipe feed rate
- Drill pipe push force
- Drill pipe torque
- Water flow rate

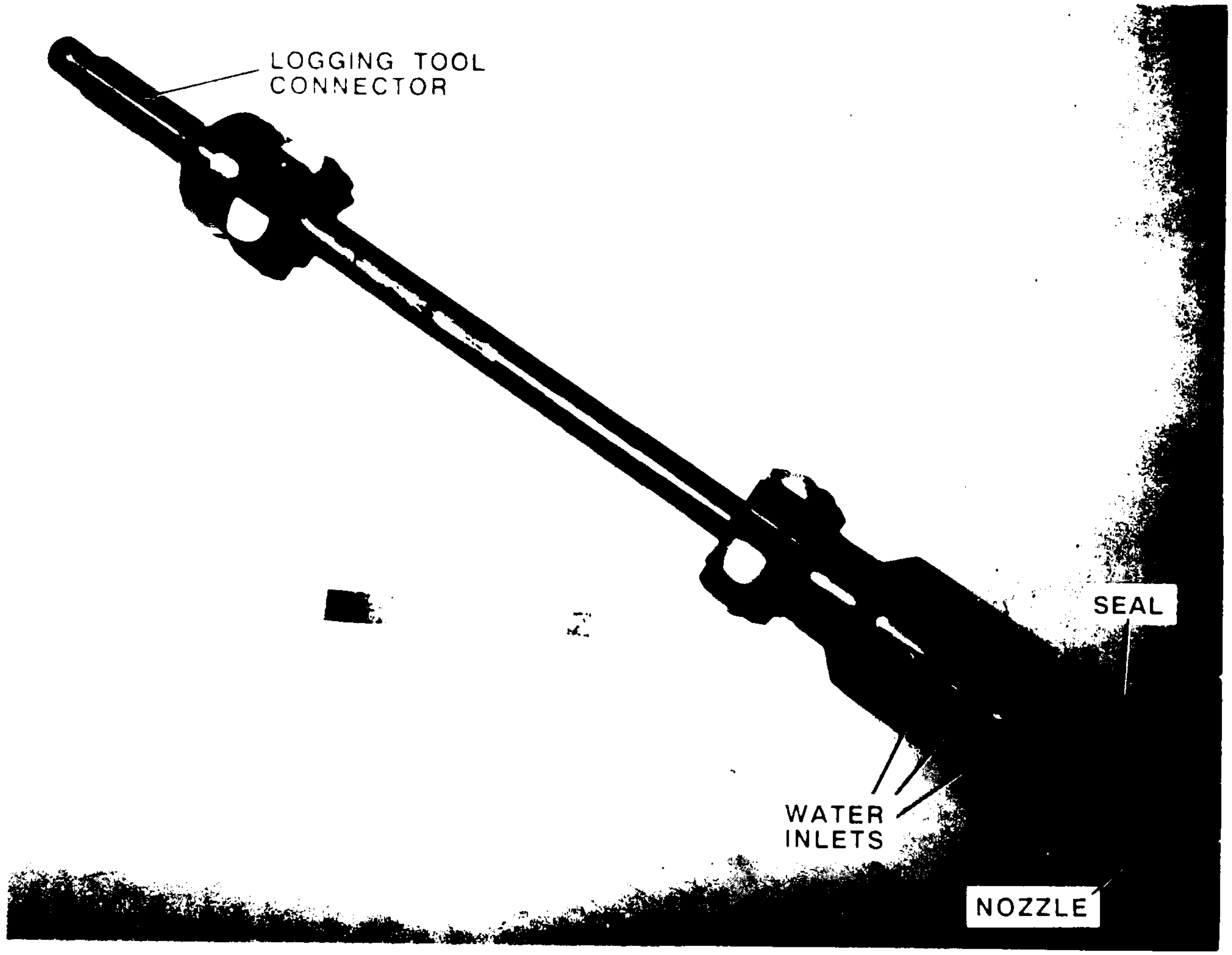


Figure 7
Straight ahead drillhead.

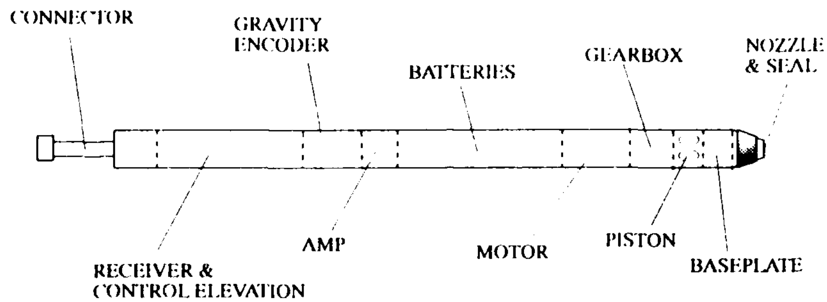


Figure 8
Steerable drillhead schematic.

During straight-ahead drilling, the operator is concerned principally with maintaining the optimum drill hole size for maximum possible feed rate and steady steering with minimum required torque. The balance required is between a large hole, which requires a lot of power and time (slow advance) and tends to cause the drill string to steer downward, and a small hole, which is faster but can lead to clogging of the cuttings, high back pressure, and upward steering.

The proposed method of maintaining optimum drilling performance is to monitor drilling water pressure and compare it with the theoretical pressure for the engine speed and associated pipe losses. The difference will be the system back pressure, assuming no equipment degradation.

The operator must consider the effects of reducing water flow rate on the measured back pressure. The operator must simultaneously measure the torque and push force as drilling indicators. When drilling for distance, all efforts must be made to minimize these values as they will be the limiting factors. That is, under most foreseeable conditions, the limiting distance factor is mechanical failure of the pipe either in torsion at the joint or in buckling. These are described in detail in Section 5.

In summary, the operations goal is to achieve the desired drilling distance, within the allowed accuracy, with acceptable system and personnel safety and to do so at maximum speed to reduce costs. This is all achieved by providing the maximum feed rate at the minimum torque, push force, and back pressure by manipulating pressure, flow rate, and orifice configuration.

4.4.1 Selecting the Drillhead Nozzle Configuration. The selection of the drillhead nozzle configuration depends upon two major considerations: the anticipated geological formations to be encountered, and the launcher machinery setup. The ideal drilling mode operation uses only one of the two pumps, at minimal pressure and flow rate, and at the maximum possible push rate.

Initially, the drillhead orifice will be sized and angled to cut the proper hole size based upon the compression strength of the geological formation. This configuration will directly determine feed rate.

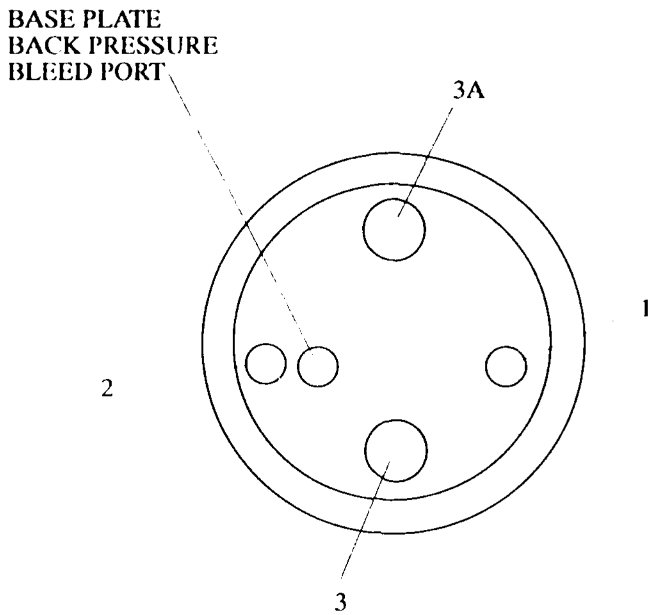


Figure 9
Nozzle orifice positions.

What is unknown at this point, however, is the effect of the geology's permeability on back pressure. In other words, if the formation cannot readily dissipate the water pumped in, the effective cutting pressure will be reduced. This may affect nozzle selection.

Tables 2, 3, and 4 and Figure 9 show the various available orifice inserts and nozzle configurations. The following relationship between flow and pressure for each of these orifices is derived from Bernoulli's equation:

$$Q = CA(2gh)^{1/2}$$

where Q = flow (ft^3/sec), A = area (ft^2), g = 32.2 ft/sec^2, h = pressure drop (ft), and C is a coefficient. C has been determined by testing to be 0.86 for the straight orifices used in the HDS.

The flow rate available with the large plungers in the pumps (3.25-inch diameter, 7.938-inch stroke) is 240 gpm. The flow rate with the small plungers (2.75-inch diameter) is 200 gpm. The orifices are sized by determining what pressure drop is desired (which determines the velocity of the water, which controls the cutting power) and then sizing the orifices so that their total area will produce flow that matches what is available from the pump arrangement in use.

For example, for a 10,000-psi nozzle arrangement operating at 240 gpm (1,400 horsepower, slightly below maximum), orifices were selected as follows:

1 = 0.100 inches (5-degree angle)
2 = 0.174 inches (15-degree angle)
3 = 0.203 inches (20-degree angle)
3A = 0.203 inches (27-degree angle)

Flow through number 1 is 0.58 ft^3/sec, or 26 gpm. Flow through number 2 is 78 gpm and flow through 3 and 3A is 107 gpm (note that 3 and 3A are not opened at the same time). That is a total of 211 gpm. It is set slightly less than the pump capacity to allow for loss in the SDH and to allow for some adjustment as the orifices wear and enlarge.

Using this procedure, orifice combinations were developed for five basic nozzle cases:

Case	Power Source	Flow (gpm)	Nozzle (psi)
1	Two diesels	200	15,000
2	Two diesels	200	12,500
3	Two diesels	200	10,000
4	One diesel	150	15,000
5	One diesel	150	12,500

The orifices selected for these cases are as follows:

Case	Small Orifice (in.)	Medium Orifice (in.)	Large Orifice (in.)
1	0.086	0.141	0.171
2	0.090	0.147	0.179
3	0.095	0.163	0.189
4	0.061	0.099	0.121
5	0.063	0.104	

In laboratory testing it was determined that the optimum nozzle arrangement for cutting granite has slightly larger orifices than even Case 3. The orifices were:

1 = 0.141 inches (7.5-degree out angle)
2 = 0.155 inches (16-degree out angle)
3 = 0.224 inches (25-degree out angle)
3A = 0.224 inches (35-degree out angle)

4.4.2 Straight-Ahead Drillhead. The straight-ahead drillhead is utilized when it is anticipated that no deviation (course change) will be required. This unit is far less complex and expensive than the steerable drillhead, and can hopefully be used for the major part of the drilling. This should greatly reduce the wear and tear on the more complicated and expensive SDH.

4.4.2.1 Nozzle Configuration. The straight-ahead drillhead incorporates four nozzles, three of which are opened at any one time. An example nozzle set is identified in Table 2 below:

Table 2
Typical Nozzle Set Parameters

Orifice No.	Angle from Horizontal (deg)	Direction	Diameter of Hole (in.)
1	13	Inward	0.086
2	7	Outward	0.141
3	18	Outward	0.171
3A	23	Outward	0.171

Orifice numbers 1 and 2 are always open, and they are the centermost cutters. Either orifice number 3 or 3A is closed off for a particular drilling operation. Testing has shown that the harder the material being drilled, the greater the outermost angle required; thus, orifice number 3A should be closed when drilling in relatively soft material, while a hard material would dictate that orifice number 3 be closed.

Three variations of the basic nozzle configuration have been manufactured. The variant in these designs is the angle of the orifices shown in Table 3 below:

Table 3
Wide, Medium, and Narrow Nozzle Configurations

Nozzle Type	Orifice Angles (deg)			
	No. 1	No. 2	No. 3	No. 3A
"Wide"	-20	13	22	33
"Medium"	-20	10	18	27
"Narrow"	-15	8	14	23

Each of these variations can be equipped with suitably sized orifice inserts. Four different orifice insert sets have been manufactured, which are identified in Table 4 below:

Table 4
Existing Orifice Insert Sets for Different Operations

Type of Operation	Orifice Diameter (in.)			
	No. 1	No. 2	No. 3	No. 3A
Single pump (15,000-psi back pressure)	0.086	0.141	0.171	0.171
Single pump (3,000-psi back pressure)	0.088	0.145	0.176	0.176
Dual pump (1,500-psi back pressure)	0.048	0.078	0.140	0.140
Dual pump (3,000-psi back pressure)	0.049	0.080	0.144	0.144

4.4.3 Steerable Drillhead. This section describes the SDH design and operation.

4.4.3.1 Drillhead Operation. The steerable drillhead is used for the controlled deviation of the drill pipe string. Turning may be up/down, port/starboard, or at any desired angle from the current drilling center line. The steerable drillhead is used in conjunction with the logging tool (presented in Section 4.7), which communicates the electronic commands necessary to instruct the steerable drillhead.

The steerable drillhead is made up of an electrical and mechanical section, each of which is contained inside a 12-foot-long by 3.125-inch-OD titanium housing.

When used in the steerable mode, a piston is shifted twice on each revolution to direct the flow to either orifice number 3 or 3A, the two outermost of the four orifices. The gravity encoder in the electrical section is used to continuously identify the rotational position of the drillhead relative to the earth.

During each revolution of the drillhead, the piston is shifted from the orifice number 3 to the number 3A position and back again. Instructions for the steerable drillhead are delivered by the logging tool or by the rotation rate of the drill string.

4.4.3.2 Drillhead Design (Figure 8). Starting from the aft end of the steerable drillhead, the first section is the mechanical connector and telemetry receiver unit. The cylindrical stab at the end includes a groove, which mates with the quick disconnect coupling on the end of the logging tool in order to complete the mechanical connection.

When deploying the steerable drillhead, this mechanical connection is made at the entrance and disconnected once the logging tool and steerable drillhead reach the end of the drill string. When the drillhead must be retrieved, the logging tool is pigged downhole with the coupling open. The coupling is then latched electrically by the logging tool after it has bottomed out and swallowed the cylindrical stab.

Within the cylindrical stab is the telemetry receiver. The housing for this end piece has four small standoffs on the outside diameter which serve as centralizers.

The next SDH section contains the circuit boards, which provide the control logic for the steerable drillhead, and the gravity pendulum and its encoder. These last two items establish a reference to the vertical position.

After the gravity encoder comes the amplifier housing, mounting board, and the battery section. The batteries are divided into a control and a power section.

In the final (front end) section are, in order, the motor, gearbox, gearbox-mounted encoder, coupling, pressure seals, piston, baseplate, and nozzle.

4.4.3.3 Instructional Steering Modes. When the logging tool reaches the steerable drillhead, located at the end of a logging run, the telemetry system in the logging tool transmits data though the acoustic coupler (located at the aft end of the steerable drillhead). The steerable drillhead can be placed into one of four distinct modes:

a. Timed deviation with an instructed direction.

b. Continual, untimed deviation with an instructed direction.

c. Roll control mode.

d. No deviational instruction but a signal to place the nozzle in either the wide cut or narrow cut mode.

Each of these four different modes is discussed below:

a. Timed Deviation with an Instructed Direction:

In this mode, the operator selects one of four preset time values. These four values can be altered prior to sending the steerable drillhead downhole by means of dip switches on the circuit board in the electronics section of the drillhead. The operator's instructional sequence in this case is:

1) Reset signal
2) Set signal
3) Timer mode signal
4) Time amount selection

In the timed deviation mode, the timer only commences when the drill pipe rotation is greater than 5 rpm. The operational scenario calls for reducing the pipe rotation speed to 4.5 rpm whenever logging operations are to be performed or whenever the forward rotator is in control of rotation (i.e., when a new pipe section is being added).

Thus, when the logging tool reaches the drillhead at the end of the pipe, the drill pipe is rotating at 4.5 rpm. The operator gives the above instructions, but the piston does not begin to shift and the timer does not start. At this point in the operation, the water pressure is low to maintain a slight positive flow into the hole.

After instructing the drillhead, the logging tool is retracted out of the hole, while the drill pipe rotation is maintained at 4.5 rpm. When the logging tool has been removed and the launcher and pipe are ready to recommence drilling, the drill pipe rotation is increased above 5 rpm and the timer begins and shifting starts. Reducing the drill rotation below 5 rpm at any time before the timer times out causes the timer to pause until the rotation is once again above 5 rpm.

When the timer's set time expires, the piston assumes a preset position of either wide cut or narrow cut. This default position is set on the circuit board, not by the telemetry unit. The maximum attainable drill pipe rotation is 9.2 rpm. Table 5 summarizes the timed deviation mode of operation:

Table 5
Timed Deviation Mode of Operation

Pipe Rotation (rpm)	Mode	Timer	Piston Motion
0 to 4.5	Timer OFF	OFF	Stopped in default position
0 to 4.5	Timer ON	OFF	Stopped in default position
5 to 9.2	Timer ON	ON	Shifting for deviation
0 to 9.2	Timer OFF	Expired	Stopped in default position

b. Untimed Continual Deviation Mode:

In this mode, deviation is stopped below 5 rpm on the pipe and the same response occurs in the two rpm zones. Table 6 summarizes the modes:

Table 6
Untimed Continual Deviation Mode

Pipe Rotation (rpm)	Mode	Timer	Piston Motion
0 to 5	Deviation	OFF	Stopped in default position
5 to 9.2	Deviation	OFF	Shifting for deviation

The operator enters this mode with the following instructions given through the logging tool telemetry unit: Reset, Set, Direction Selection.

c. Roll Control Mode:

In this mode of operation the timer is not used. The function of the drillhead is governed strictly by the drill pipe rotation speed. Table 7 summarizes the performance of the drillhead in this mode:

Table 7
Roll Control Mode

Pipe Rotation (rpm)	Status
0 to 5.5	No deviation-default position
5.5 to 6.5	Up deviation
6.5 to 7.5	No deviation-wide cut
7.5 to 8.5	No deviation-narrow cut
8.5 to 9.2	Down deviation

Note that this assumes steady rotation of the drill pipe end. No tests were conducted to investigate the effects of unsteady rotation.

d. No Action Mode:

A final choice available to the operator is: after cycling the logging tool downhole, make no changes and leave the drillhead in its defaulted condition.

4.4.3.4 Steering the Drillhead. Steering the drillhead is required to achieve controlled directional guidance. This new horizontal drilling mode, however, does not have the benefits of conventional rotational drilling control methods; there is normally no direct contact between the working element (drillhead) and the geological formation, and there is no casing or annulus around the pipe.

If all four of the orifices were left open as the pipe rotated, the cutting pattern would be as shown in Figure 10a for a straight and narrow cut, and as shown in Figure 10b for a straight and wide cut. The cutting pattern would appear in front of the nozzles and the rings would indicate the individual orifice cutting zones.

When an UP cut is selected, the piston is shifted each revolution and the pattern shown in Figure 10c results.

When orifice numbers 3 and 3A are UP, they are in the open position; when they are DOWN, they are closed.

This ring pattern is the same for any selected direction. It is merely rotated by changing the piston's shifting instructions.

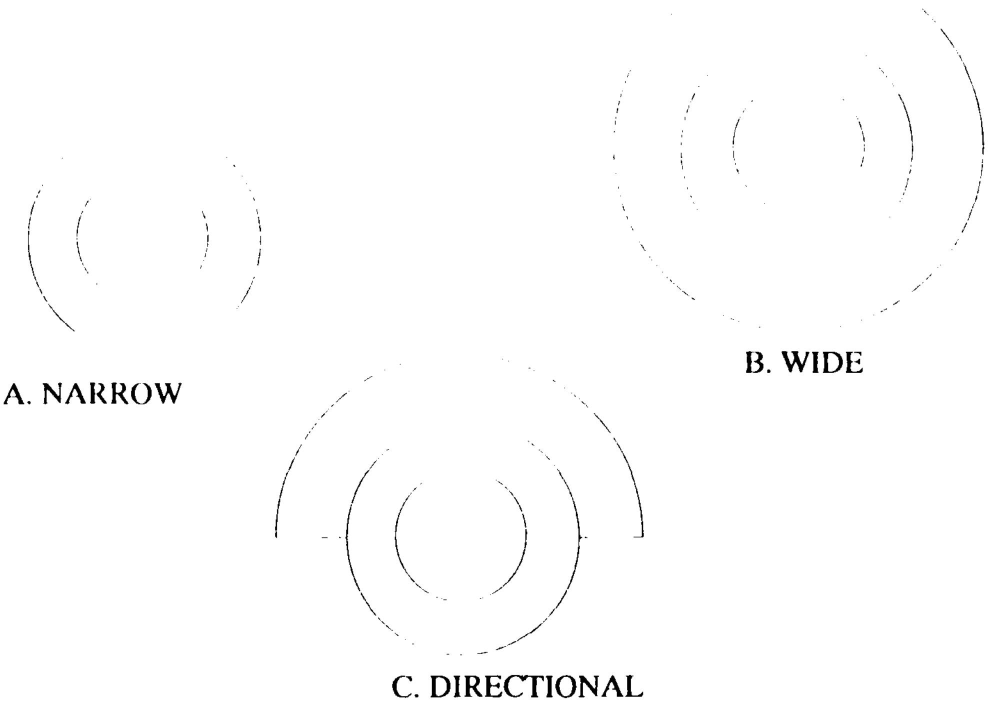


Figure 10
Cutting patterns.

Two theories exist as to how the drillhead is actually deviated. Both theories may have validity:

a. Theory No. 1. The SDH controls direction by creating a hole that is elliptical and offset from the center line of the pipe. This action forces physical contact between the rock or soil and the tapered nose of the drill pipe on the side away from the direction of the cut. The resulting contact pushes (or deviates) the pipe to one side and a turn is initiated.

Since this "turning" method depends upon physical contact between the pipe and the formation, the deviated hole size is critical. Likewise, the ability to turn greatly depends upon the geological formation encountered, and may require considerable field testing to be able to predict turning operations with any degree of accuracy (particularly in soft formations). It is anticipated that drillhead "turning" performance can be observed by monitoring the drill push and torque forces. If this theory is correct, then a very small deviation time will be needed to "kick off" the drill string.

b. Theory No. 2. Under this second theory, the end of the drill string does not contact the hole. An elliptical hole is cut and it is big enough to allow the drillhead to pass. Deviation occurs because the cutting buildup against the pipe walls is tighter on the narrow side. Thus, the pipe is gradually deviated. Under this theory, steering takes longer because the deviation is more gradual.

4.5 Drill String as Casing

One major advantage to the HDS design is that the drill string is used as the final casing. This saves a lot of time and expense because as soon as the hole is drilled, the system is ready for cable installation. Of course, it may be desirable to case the entrance and the first few feet of the hole to help stabilize that section against hole collapse. It is also important to establish this first casing in the event there is any possibility of encountering high pressure fluids while drilling. The casing (sealed in place) would be required for the installation of a blowout preventer or other seal if required. The initial geological surveys should provide the information needed to make the decision regarding use of a seal on the pipe at the entrance. In general, it is unlikely that it will be needed.

Leaving the drill string in place as a casing places some limitations on the drill string diameter. It has to be large enough internally to contain one or more cables. The present design will easily contain six to eight 1/2-inch cables, or two SD List 1 cables (1.25-inch diameter) or other such combinations. The only limitations are those encountered in pigging the cables (difference between pipe ID and cable total cross section). The sliding seals on the cables will probably not allow extremely high pigging pressures. Tests have not been performed to explore the limits but general experience indicates that a gland seal on the cable will not likely function well much above 2,000 to 3,000 psi. The theory for calculating required pull forces on cables is contained in Reference 3. However, as an example, the estimated pull force for a single 1/2-inch cable at 25,000 feet is 1,104 pounds. For six such cables the required force would be 6,624 pounds. Their total cross-sectional area is 1.30 inches. Subtracting that from the pipe interior cross section leaves a net of $(9.62 - 1.30) = 8.32\ \text{in.}^2$. The required pigging pressure would be 796 psi (plus enough to overcome pigging seal friction), which is well within the seal limits. If the net cross section becomes too small to produce enough pull force to pig cables at the maximum pressure, the alternative is to pig a messenger through and then pull the cables mechanically. Note that cables may be pulled from either end. It is normally preferable to send them from shore because it reduces the time on station for the ship at sea.

The pipe also must have enough wall thickness to survive hole collapse, corrosion, or anchor strikes (at the exit point) for several decades. The present 0.625-inch wall is more than ample for all these effects.

Use of the pipe as a carrier for electrical current is also a possibility. The pipe may become part of a corrosion control circuit and it may carry high currents in the event of High Altitude Electromagnetic Pulse (HEMP) effects from nuclear attack. The pipe also may have a tone applied from shore for use in locating the pipe at sea or surveying its position. The pipe joints are not presently designed to guarantee low-resistance electrical contact across the joint over long periods of time. If this feature is desired it may be possible to enhance the electrical connectivity by using conductive grease or other additives at the joint.

4.6 Improved Joint Design

The improved joint design is a key technical feature of the HDS. References 4, 5, and 6 describe the joint development. Details for assembly and care of the joints are provided in the Operations and Maintenance Manual (Ref 2). Section 5 of this report analyzes the effect of the joint's stress limitations on the operation of the drilling system. The important point to remember here is that the joint is the likely ultimate weak link in the system unless proper controls are maintained at the launcher to prevent joint failure. The overall purpose of this

operations theory is to provide a method to plan the hole, set up the SDH or SADH, and operate the launcher in a way that doesn't fail the joints before the desired hole length is reached.

4.7 Supporting Equipment and Methods

In addition to the main drilling system (power sources, water supply, launcher, pipe, and drill heads), there are supporting instrumentation systems and methods that must be used to monitor the progress of the drilling operation. Some of these systems are stand-alone (such as the logging system) and others are built into the main drilling system (the instrumentation control console and its distributed sensors). There are of course major supporting references providing the "how to" information regarding these systems. However, it is important to consider their general role in the overall operation as part of the basic operations theory.

4.7.1 Logging Subsystem. A horizontal drilling system requires that the hole be drilled to a specific point with a reasonable degree of accuracy. Therefore, the system used to measure the location of the hole is a major stand-alone subsystem.

The device used to determine downhole positioning is called the logging tool (LT). From the collected LT data, the drilling operator can then both equip and control the main drilling machinery to accomplish the necessary directional guidance. Figure 11 shows the LT schematic.

While logging is an everyday occurrence in the commercial oil industry, HDS requirements are slightly more complicated. The drill pipe is smaller than normal and is constantly rotating. The LT also must communicate with and instruct the steerable drillhead without loss of rotation or water flow.

The logging tool has three primary sensors: two for determining elevation and one for azimuth. Elevation changes are measured by Sundstrand accelerometers mounted on a de-roll platform (i.e., the accelerometers are held level as the pipe rotates around them). A second verification of elevation is given by a low pressure transducer, which measures "static" (low velocity flow) water head in the pipe.

This transducer has an automatic protection device that guards it from being exposed to pressures above 500 psi. When water flow is reduced to a trickle in the pipe, the transducer gives a very accurate depth reading.

The requirements for accuracy of the depth measurement are much tighter than the requirements for horizontal positioning. That is because the exit onto the seafloor must usually be performed on a very gently sloping seafloor; small errors in the knowledge of pipe depth could produce large errors in the location of the exit point. Although the allowable depth target is generally about ± 15 feet (to stay in the 60- to 90-foot diver depth range), the knowledge of the depth must be considerably greater than that. As a general rule, it is necessary to measure to significantly greater accuracy than the required error in actual control. In this case, a reasonable requirement for measurement accuracy would be about 10 percent of the required control accuracy, or ± 1.5 feet.

For example, if the overall measurement range of the instrument is a maximum of 350 feet of elevation difference between the launcher and the maximum depth of the pipe, then the required accuracy is $(1.5/350) = 0.4$ percent (not totally unrealistic but a stringent requirement). There are three main sources of error in the reading. One is the error in the instrument itself, the second is the fraccing pressure, and the third is the error in the knowledge of pressure drop from flow loss due to the water that must remain moving through the pipe to prevent back-flushing of contaminants into the SDH. Assuming the errors are independent and equal, that

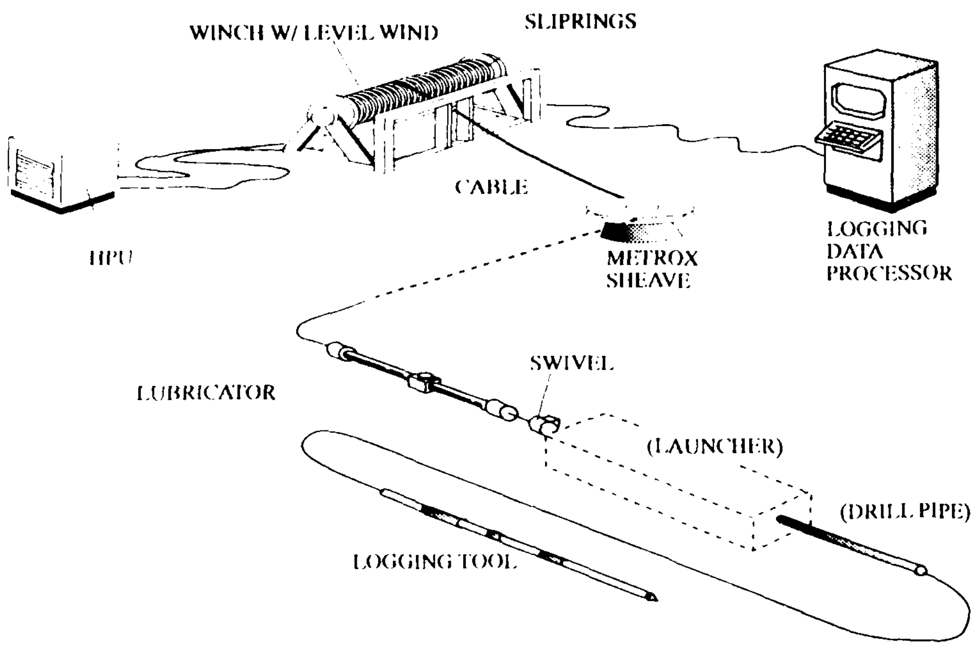


Figure 11
Logging tool system.

would mean the allowable error from each source is ±1.02 foot. From Bernoulli's equation for flow loss in a pipe from flow Q,

$$h_Q = fLV^2/2gD_i$$

and for the existing pipe dimensions,

$$h_Q = 14.64\ V^2$$

Thus, if the allowed head loss error is < 1.02 feet,

$$V_{max} = 0.269\ \text{ft/sec}$$

which is a flow of 8 gallons per minute.

Therefore, even with a pressure gauge that is accurate to ±1 foot, the flow must be less than (or known to within) 8 gpm to ensure an overall accuracy of ±1.5 feet. The largest source of error will be fraccing pressure (stored in the formation). One way to check this is to time the pressure - fraccing pressure will decay with time, static head will not.

Azimuth is measured by a caged rate gyro which is also mounted on the de-roll platform.

Random error is reduced by checking the data on the first part of the hole each time a new logging run is made. It is intended that systematic error elimination and calibration will be performed during field test operations. The logging tool will be run down a number of known paths until calibration and operation are sufficiently understood.

4.7.2 Launcher Operations to Limit Pipe Loads. The data from the sensors of the information control center provide the operator with status of the launcher and pumping equipment. They also show the primary loading on the pipe at the launcher end. Fortunately, theory indicates that in nearly all cases the maximum loading on the pipe or joints will be at the launcher end. That is good, because the operator really has very little direct, real-time information about what is happening down at the drillhead end of the string. The operator must rely on the theory developed in this report and the experience gained in testing to relate what he can measure at the launcher end to what is happening at the drillhead in real time. That information is supplemented by logging data and visual inspection of the drillhead whenever it is retrieved for change out. The actual operation of the system still must be based largely on what can be seen, heard, and measured at the launcher control console. The theory behind the following discussions is developed in Section 5. The basic conclusions are presented here in advance to provide an overview of the key functions of the system.

4.7.2.1 Starting the Hole. Buckling of the drill string could potentially be caused by sudden loss of drilling pressure while pushing on the drill string with high force. Redundant automatic protection controls (described in Appendix A of Reference 6) and operations procedures (described in the Operations and Maintenance Manual, Reference 2) have been incorporated to minimize this potential.

This section deals with another potential source of buckling - the creation of unacceptably long unsupported spans of drill pipe in the hole.

The potential for pipe buckling is one of the most critical factors that limits drilling distances. This unconventional type of drilling operation has increased the pipe resistance to buckling in two ways. First, the principal of using high internal water pressure places the pipe in tension (initially). The pipe only begins to be subjected to buckling forces when the push force required to advance the pipe in the hole exceeds the pretension induced by the internal pressure. Second, the unsupported length of the pipe outside of the hole has been reduced by the addition of intermediate supports to a maximum distance of 4 feet between supports. At the onset of drilling, it is very important to avoid the creation of relatively large voids in the hole which result in excessive unsupported pipe lengths. The buckling forces on the pipe are greatest near the launcher and diminish progressively downhole.

At the start of drilling, the recommended nozzle configuration is a narrow angle nozzle with inserts configured for 15,000-psi pressure with only one pump running. It is important not to create a bigger entrance hole than is required to pass the pipe. The exact nozzle and angle arrangement required will vary with soil conditions.

For example, in soft material the above arrangement may result in too large a hole. In this case, the operator can reduce engine rpm and use the choke to limit pressure and flow to minimize hole size. Conversely, in harder material like granite, a two-pump, wide angle arrangement may be required to start the hole.

Another consideration at the start of drilling is to prevent hole washout at the entry point. Basically, the cuttings bed which supports the pipe downhole should extend all the way to the hole entry point. In the early stages of drilling, the penetration rate should be sacrificed to

ensure a hole shape that supports the pipe adequately. This is best accomplished by a reduced flow rate to avoid washouts and by narrow angle nozzles. Sandbags or other types of barriers at the entry point to prevent cuttings washout may be required.

4.7.2.2 Controlling the Advance Rate. While economics dictate use of the most rapid advance rate possible, the achievable drilling distance can be seriously diminished by an overly aggressive advance rate. The limits of reach with this drilling rig have been calculated using an assumed coefficient of friction between the pipe and the geologic material. The calculations are based on pipe weight, steering forces, and the resultant friction forces.

Significantly higher resistance is generated by trying to shove the pipe into a hole which is cut somewhat smaller than the diameter of the pipe. This condition creates binding and bending and increases the torque and push force requirements. If a small projection of hard material remains in contact with the pipe after the cutting nozzle has passed, it may cause an unwanted deviation and increase the resistance.

The way to maximize the reach of this drilling system is to drill as straight as possible for as much of the hole as possible and restrict the deviations to very short, planned locations and amounts.

The operator should know that drilling too fast will cause problems and that drilling slower than required is economically unacceptable. The operator cannot see what is going on at the drillhead and he does not have conventional feedback such as weight-on-bit and real cutting rate.

The advance rate relates to buckling. If the advance rate is too high and unwanted deviations occur, then the onset of buckling will start at reduced drill string distances.

Controlling the drill penetration rate is critical from two points of view: (1) to ensure the maximum drilling rate possible, and (2) to ensure against drill pipe overloading conditions (and potential breakage). Drill penetration rate can be regulated by managing three principal drilling parameters:

a. Drill Push Force
b. Drill Rotational Torque
c. Drill Water Pressure

Different scenarios involving these parameters are presented below for review:

a. Drill Push Force. Launcher hydraulic controls allow the operator to adjust the available drill push force created by the large cylinders. In order to ensure forward progress, the baseline aproach is to set the push force (pressure reducing valve) at a slightly higher level than the nominal resisting force.

During the initial phase of drilling, with very little pipe in the hole, the resistance will be very small and the (cylinder) pressure reducing valve should be set just high enough to advance the pipe. The benefit of this operating mode is that if the operator tries to advance the pipe faster than the drillhead is cutting, the pipe will just stall out without excessive buckling forces being induced on the pipe. A second benefit is that the operator should get some indication when unwanted deviation is occurring.

b. Drill Rotational Torque. This control is very similar to the push force control. The operator has a pressure reducing valve that controls the rotational torque applied to the pipe. The

same approach applied to the push force method applies in this case. The hydraulic pressure (and hence torque) is set as low as possible to maintain rotation.

An indication of contact at the drillhead will be seen as a rise in torque. The operator observes this as a rise in pressure or as a slow down or stalling of the pipe rotation. When the operator observes pressure rise, slow down, or stalling, the following should be performed immediately:

1. Increase rotational pressure if trying to deviate or if it is suspected that the drag is normal and due to the increased penetration rate.

2. Stop advance, back up a short distance and redrill the same place if in the straight-ahead mode and adequate torque is already applied.

c. Drill Water Pressure. Another variable the operator can observe is the system water pressure produced by the large triplex pump(s). Each pump is a fixed displacement type so flow is proportional to engine speed (1,250 rpm on one pump equals approximately 100 gpm). Use of the second pump simultaneously doubles the flow.

During drilling, the operator observes the water pressure and engine(s) speed(s). The operator is provided with a curve which shows the expected water pressure versus engine speed. If the observed pressure exceeds the predicted amount, the difference is back pressure. By subtracting the back pressure from the observed pressure, the operator calculates the pressure drop across the nozzles. This provides an estimate of the available cutting power.

As an example of this method, suppose the drilling is proceeding at 1 ft/min in difficult material. Both pumps are running at 1,200 rpm and a nozzle set designed for two pumps and 1,500-psi back pressure is in place. The drilling pressure is 14,000 psi. Now, suppose the drilling pressure gradually increases so the operator must reduce engine speed (flow) in order to keep the drilling pressure under 15,000 psi.

Eventually, the operator reaches 1,100 rpm on both pumps. This is the lower speed limit for full load operation on the diesels. The penetration rate has dropped to 1/4 ft/min. Something must be changed.

The problem is either: (a) a plugged nozzle orifice (unlikely, since the change has occurred gradually), or (b) the drilling is progressing into very nonpermeable geology and the back pressure has increased.

Assuming that permeability is the problem, the operator may:

1. Install a small choke orifice and open the choke, and try to penetrate even though penetration rate is very low.

2. Replace the drillhead with a nozzle configured for one pump; this will cut the flow in half.

Another example assumes the same conditions as in the previous example but now the drilling pressure goes down. There are two possibilities, either the nozzle insert is wearing (becoming enlarged) or the pump performance is degrading.

The operator can verify that the pump is working by installing the 12/64-inch orifice into the choke, opening the choke, closing the 3-inch valve, and calibrating the pump. If the pump

pressure versus engine speed matches the new condition (and the speed of the engine is correct), then the cause is nozzle wear or a leak path such as the rubber seal on the end of the SDH.

4.7.2.3 Preventing Drill Pipe Failures. Mechanical strength of the drill pipe or joint is expected to be the limiting factor. A drill pipe breakdown may result from pipe buckling, stresses due to torsion, or wear-induced failure.

The limited mechanical strength of the drill pipe complicates system operation and controls. The internal pressure caused by the high pressure water puts the pipe in tension. During the drilling mode, both the pipe length and push force (needed to overcome friction) will increase to the point where it exceeds the initial tension, and the pipe will be under steadily increasing axial compression at the launcher end of the drill string.

As this pipe axial compression increases, the likelihood of a buckling failure increases. Buckling can occur in a section of the hole that is soft or large (that is, caused or blown out by water-jet drilling), which results in unsupported lengths of pipe.

As frictional forces increase during drilling, the force required to turn the pipe will also increase. While this failure mode is easier to predict, the maximum amount of torque that can be transmitted through the pipe joints is limited by the yield strength of the joint. While push force is positively affected by the tension created in the joint (by the internal pressure), torsional capacity is negatively affected. That is, the greater the internal pressure, the lower the pipe joint capacity. In addition, the combination of internal pressure and thrust force reduces the torque capacity of the pipe below levels due to pure torsion.

Numerous safety devices and computer-controlled valves limit "operator controlled" operation at certain times, ensure push/torque/pressure forces increase proportionally, and activate other safety features. While these features only come into play after a considerable length of hole is drilled, they add to system complexity. These controls are explained in detail in Appendix A of Reference 6. <u>NOTE: The control equations have not been updated to reflect recent test data, and will need to be modified before further use</u>.

In addition to the automatic control features that prevent the operator from entering into destructive combinations of pressure, torque, and push force, the operations procedure has been modified to prevent destructive combinations. In this way, a measure of redundant protection has been achieved. Section 5 provides the details of the analysis supporting these observations, and works through a sample problem to illustrate how the various HDS subsystems work together.

5.0 ANALYSIS OF DRILL PIPE STRESSES VERSUS DRILLING OPERATIONS

As discussed in Sections 3 and 4, even when all supporting systems are working properly and the local environment is favorable, the accumulated stresses on the drill pipe will eventually set limits on the total distance and route that HDS technology can achieve. This section of the Operations Theory Report provides a simplified mathematical model of the loads that are expected on the pipe as a function of the various drilling parameters. Those loads are then related to the resultant stresses through conventional engineering models. The derived relationships may then be used to develop guidance for:

a. Planning the drill route.

b. Controlling operational equipment settings.

c. Interpreting drill string behavior during drilling.

d. Making adjustments as required to produce the best installation that site conditions, drilling equipment, time, and money will allow.

5.1 Definition of Terms

For purposes of the HDS analysis, the following definitions are adopted:

A_i	=	cross-sectional area of interior of pipe (in.^{2}) = 9.62 in.^{2}
A_w	=	cross-sectional area of pipe wall (in.^{2})
d	=	effective depth of soil over a collapsed hole (ft)
D_h	=	inside diameter of the drilled hole (in.)
D_i	=	inside diameter of pipe (in.) = 3.5 in.
D_o	=	outside diameter of pipe (in.) = 4.75 in.
E	=	elastic modulus (Young's Modulus) for pipe material (lb/in.^{2}) = 30 x 10^6 lb/in.^{2}
F_b	=	the force required to push/pull the pipe through the hole against friction caused by bending the pipe in a curved hole (horizontal plane) (lb)
F_D	=	force required to push the pipe (forward or reverse) (lb)
F_f	=	the force required to push/pull the pipe through the hole against friction caused by gravity forces alone (vertical plane) (lb)
F_g	=	the force required to push a pipe up or down an incline against gravity (no friction) (lb)
F_o	=	vector sum of F_r and F_D = $\mu\omega_o L$ (lb)
F_r	=	force required to rotate the pipe (at surface element) (lb)
F_R	=	tensile, or "rabbit" force created in the pipe by the application of internal pressure P (lb)
F_s	=	the net push force required to produce sinusoidal buckling of the pipe in the hole (lb)
I	=	moment of inertia of pipe cross section in bending (in.^{4})
l	=	critical length of unsupported pipe (ft). The transition between a buckling failure condition and a material failure.
L	=	length of pipe (ft)
P	=	pressure (lb/square inch)
r	=	outside radius of pipe (in.) = 2.38 in.
T	=	torque applied to the drill pipe (ft-lb)
u	=	geometric stiffness factor
V_D	=	linear forward velocity of pipe surface from drilling (forward or reverse) (in./min)
V_O	=	vector sum of V_R and V_D (in./min)
V_R	=	linear velocity of pipe surface from rotational motion = 74.6 in./min at 5 rpm
x	=	deviation of pipe from original course left or right of track, horizontally (ft)

y = horizontal distance pipe moves along the intended track (ft)

Y = launcher setback distance from beach (ft)

z = deviation of pipe vertically from the horizontal plane through the zero reference (usually at the launcher flush box) (ft)

Note that x, y, and z are a right-hand, orthogonal set of axes that form the primary survey reference frame for drilling operations. These axes must be coordinated with a local geographic reference frame for offshore operations, interface with geologic surveys, etc.

Z = elevation of launcher (ft)

α = angle between the rotational velocity vector and forward motion velocity vector at the surface of the pipe (deg)

β = net change in heading from a horizontal steering event (deg)

γ = local slope of hole (deviation from horizontal) (deg)

Γ = net change in dip angle (climb or dive) after a vertical steering event (deg). Note that $\Gamma = 2 \times \Theta$ for upward steering and $\Gamma = \Theta$ for downward steering.

Θ = bend angle of pipe at start and end of steering (deg)

μ = coefficient of friction between pipe and soil (dimensionless)

ρ_s = density of rock or soil on a collapsed hole (lb/ft^3)

ϕ = geometric stiffness factor

ω_b = distributed load per foot to bend pipe along a curved hole (horizontal plane) (lb/ft)

ω_g = weight per foot of pipe, flooded, without cable (lb/ft) = 31.82 lb/ft

ω_o = vector sum of ω_g and ω_b; distributed lateral loading on pipe that causes resistance to rotation or advance (lb/ft)

"section" = an individual (30-foot) length of pipe. The term "section" is used instead of the more common oil field term "joint" to avoid confusion with discussions of the actual pipe joint assembly (the threaded coupling, with friction washer and seals).

"joint" = the coupling assembly that connects two pipe sections. It consists of the threaded pin on one section end, the threaded box on the adjoining section end, the friction washer, and the seal.

5.2 Pipe and Joint Specifications

The detailed specifications for the pipe and joint are provided in Reference 4. Figure 12 is a summary of the pipe and joint design. The pipe is commercially available steel drill pipe, although it is a special product not normally in stock. The outside diameter is 4.75 inches and the inside diameter is 3.5 inches, producing a wall thickness of 0.625 inches. It was procured in sections about 30 feet long. The material is 4145 steel. The steel has a yield strength in tension of 135,000 psi. That is a relatively high strength, but not the maximum possible. This alloy was selected as a reasonable compromise between the needed high strength, ductility, machinability (for threading), corrosion resistance, and cost. Other materials could be used for future designs to provide higher strength for the final sections of pipe.

The joint design was based on the analysis of various configurations to maximize the allowable torque, given the expected internal pressure and other tensile or compressive loads. Note that the analysis of Reference 6 does NOT address the combined effects of additional

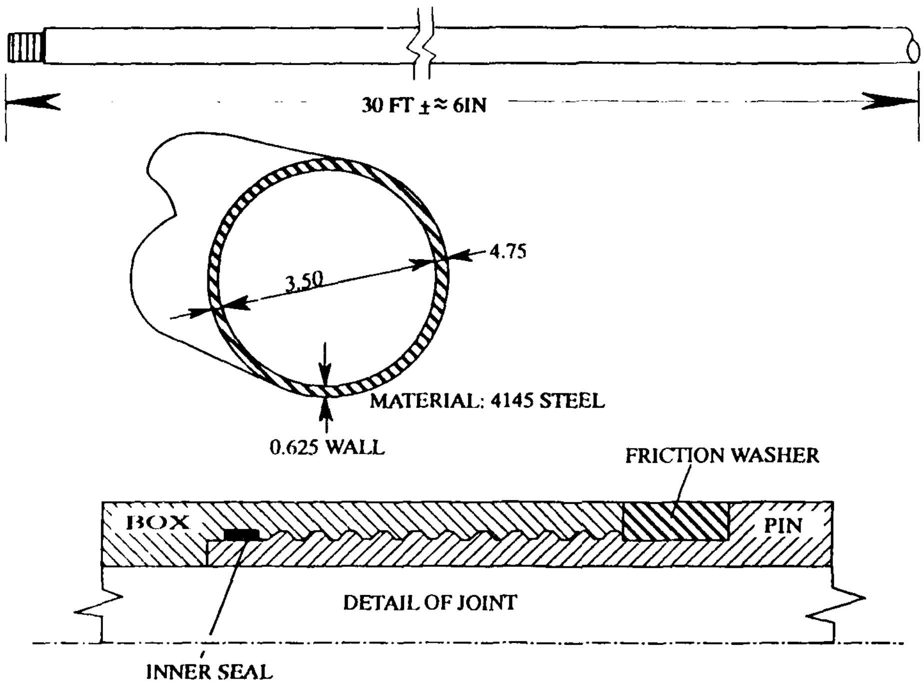


Figure 12
Pipe and joint design.

stresses from BENDING. The analysis in the following sections checks the magnitude of this effect without explicitly including it in the overall joint stress modeling effort. Tests of cyclic fatigue on the joints in the presence of bending also have not been done. These tests will need to be part of future test programs.

In general, the joint is a straight threaded pin-and-box design, with a special friction washer added between the outside shoulders of the pipe sections and a stiff radial seal around the inner end of the pin. In the absence of any pressure, tension, or compression on the pipe, the joint makes up until the friction washer locks and the seal is compressed. As torque is increased, the following happens: (1) the friction washer eventually slips due to yielding of the material adjacent to the washer, (2) the leverage of the threaded section takes effect as the pin and box rotate, and (3) the box fails in shear and compression and/or the pin fails in shear and tension near the base of the shoulder. Theory indicates that the limit of torque without compression from the friction washer, pressure, or push force is about 62,000 ft-lb. The pipe joint has been tested with the friction washer and torque has been applied only to 52,000 ft-lb without failure, but with considerable plastic deformation of the box shoulder section. For comparison, the best commercial joints for this pipe size and material would handle about 20,000 ft-lb of torque.

In the presence of internal pressure, additional tensile stress is added and concentrated on the pin, and hoop stresses are introduced throughout. The pressure also reduces the compressive thread loads on the friction washer, making that unit more likely to slip under torque. This problem can be overcome by using a relatively large makeup torque of between 20,000 and 25,000 ft-lb when installing the new pipe section. The present design produces balanced loading in nearly all modes, so stress limits occur in pin tension, box hoop tension, axial compression,

and shear at about the same loading combination. The net effect degrades the joint torque rating, so that at full internal pressure (15,000 psi), if there is no compressive load (moving forward very slowly), the theoretical limit for torque is about 37,000 ft-lb. In one test, the joint held about 35,000 ft-lb under these conditions before significant yielding was observed. The joint did not leak during the test and could be unthreaded after the test and reused if required.

In summary, the joint is the weak link in the pipe and is limited to certain combinations of torque, pressure, and push/pull force. The following analyses look at various drilling operations and estimate the effect of those operations on pipe (and thereby, joint) loading.

5.3 Analysis of Forces on Pipe/Joints

The analysis of loading and resultant stresses in the pipe is organized by the various possible drilling geometries:

a. Steering or Straight. For purposes of this analysis "steering" means any curving hole, regardless of whether the curvature is produced by commands to the steerable drillhead or through natural behavior of the drill string system. It is highly unlikely that any hole will ever be perfectly straight, so for purposes of this analysis a hole is considered straight if the effects of curvature would be too small to measure (generally much less than 1 percent). When steering does occur, it is analyzed separately as vertical steering (in the vertical plane only) or horizontal steering (horizontal plane only) and the results are added vectorially.

b. Level or Sloped. If the drill string is level, all the gravity forces (or bending forces) contribute fully to the push and rotational forces required to overcome frictional loading. If the drill string is moving upslope or downslope, there is an additional component of the gravity vector that adds to or subtracts from the push force required. This force is independent of rotational speed. In this condition, there is a slight reduction in the gravity force contribution to friction (cosine of the slope). However, for realistic slopes that effect is small in comparison to the effect of added tension/compression generated by the direct gravity vector (sine of the slope). This effect applies to both straight and curved sections of pipe. Small pipe slopes have no effect on the calculations for horizontal bending forces.

The analysis proceeds through these various geometries in increasing order of complexity.

There are four primary sources of loading on the pipe and joints. They are illustrated in Figure 13 and explained in the following:

a. Gravity-Generated Friction. The pipe weighs 27.53 lb/ft empty, dry, and in air. Flooded with salt water (but in a dry hole) it weighs 31.82 lb/ft. The pipe could weigh as much as 32.16 lb/ft when the logging cable is inside, but when that happens the internal pressure can be kept less than the maximum drilling pressure, so the joints do not see maximum stress at that time. Any flooding of the hole reduces the effective weight by up to 24 percent but it is not prudent to count on that effect over long distances. Therefore, the assumed weight of the pipe for these calculations will be:

$$\omega_g = 31.82 \text{ lb/ft}$$

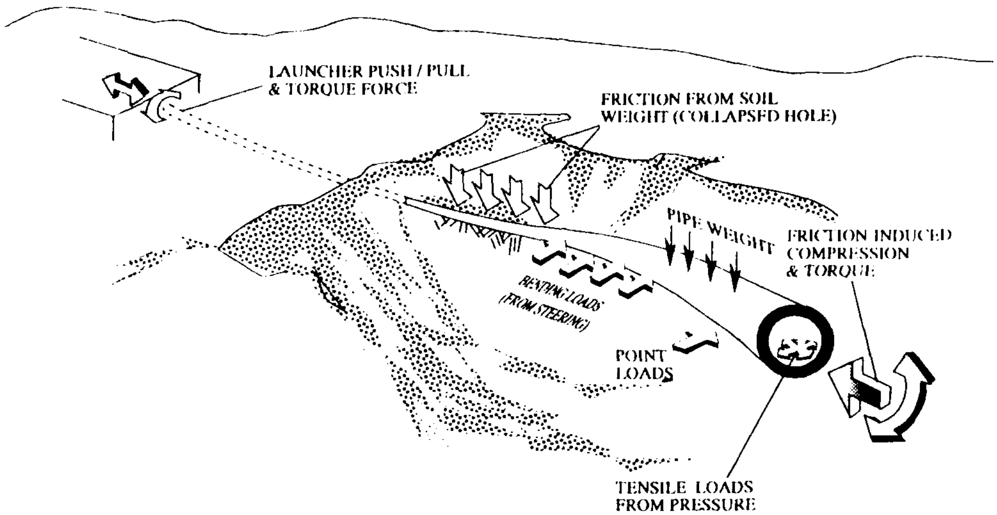


Figure 13
Sources of loading on pipe and joints.

Gravity also causes the high loading that occurs when the hole collapses around the pipe. In that case, the loading per foot is a function of the soil density and the effective depth of the collapsed soil over the pipe. This effect can be orders of magnitude greater than the pipe-weight friction in extreme cases.

In general, gravity is the dominant source of frictional loading on the pipe. It produces resistance to rotation and to forward/reverse movement of the pipe. It can aid the forward movement if there is even a slight downhill dip to the hole, but it also can retard forward movement on an uphill climb such as during the final climb to exit onto the seafloor.

b. Steering Loads. The steering loads are the loads applied to the pipe by the offset hole to cause the pipe to bend horizontally and thereby change the direction of drilling. These loads produce increased frictional loads that resist rotation and advance. They also produce bending stresses in the pipe and joint that cycle as the pipe rotates.

c. Launcher Forces. The launcher forces are the applied torque and push/pull forces at the launcher to overcome the gravity and steering frictional forces, and thereby move the pipe forward during drilling.

d. Pressure. The internal water pressure used to provide the high velocity water jet at the steerable drillhead produces high tensile and hoop-stress forces in the pipe. The tensile force (called a "rabbit force") is essential to reduce net compressive loading and prevent buckling of the pipe. However, it can lead to yielding of the joints at high torque levels. The hoop stresses can affect joint performance as well. Also, the internal pressure provides a significant geometric

stiffening effect on the pipe against bending deformation. Thus, more torque and push force are required for a given steering action with internal pressure than without it.

There are other special loads such as the compressive loads encountered if the drillhead strikes the end of the hole, wear forces from abrasion near the drillhead, gripping forces from handling tools, etc. Fortunately, these are generally very small compared to the primary loads and are not expected to limit the performance of the system. The only time they are of concern is when there is a very long period of rotation at a single position so that local wear or fatigue becomes a problem. This is mitigated by occasionally moving the pipe backward a few inches if sustained rotation at a single position is required.

The analysis of forces on the pipe and joints is conducted in two parts. The first part analyzes the forces that occur on the pipe during various drilling operations. These operations move from simple straight and level drilling, through vertical deviations to horizontal deviations, and then to the collapsed hole problem. In the second part of the analysis, the response of the pipe and joints to those forces is analyzed. The two responses considered are buckling failure (pipe) and yielding under combined loads (joint failure).

5.3.1 Straight and Level Drilling. In an open hole, for a nonrotating pipe, the equation for predicting the force required to push the pipe forward (F_D) is very simple. It is the product of the coefficient of friction between the pipe and the soil or rock (μ), the weight per foot (ω_g), and the length of the pipe (L):

$$F_D = \mu \omega_g L \text{ (lb)}$$

There is much drilling data available to estimate coefficient of friction. It is affected by the type of geology, pipe material, and presence or absence of drilling fluids. However, a reasonable number for static friction is generally about 0.48. That would imply that for 25,000 feet of pipe, weighing 31.82 lb/ft, it would take about 381,840 pounds of force to push the pipe forward. Even the simplest checks of buckling equations (see later sections) show the pipe would buckle with only a few thousand feet in the hole under these conditions. Also, the wall of the pipe pin is so thin that it would fail in compression under these conditions. Clearly, it is not possible to push a nonrotating ("static") pipe out to these distances.

The most effective change that can be made is to continuously rotate the pipe. Rotation produces two immediate improvements:

a. The coefficient of friction reduces from 0.48 to about 0.33 (a 31 percent improvement) under dynamic conditions where the sliding movement is continuous. This effect has been demonstrated in laboratory tests and is described in Reference 7. Field testing may refine this number, but not dramatically.

b. More importantly, forward motion of the pipe now becomes a relatively minor shift in direction of the net motion vector. According to the principles of engineering mechanics, as long as the pipe is rotating, even a very small push force will result in a forward motion of the pipe. It is primarily this effect that allows the HDS to reach long distances without buckling the pipe. Of course, there are additional mechanical and operational costs to maintain rotation but such costs are balanced by improved performance. This rotational vector effect is described in more detail later in this section.

The second change that can be made to improve the drilling length limitation is to add internal pressure to the pipe. The internal pressure provides a net tension (or "rabbit" force) in the pipe that overcomes much of the compressive load and greatly extends the level of total push force that can be applied before the net compressive stress reaches buckling limits. The tensile force (F_R) generated by the maximum internal pressure of 15,000 psi is equal to the product of the pressure (P) and the pipe internal cross-sectional area (A_i), which is 9.621 square inches:

$$F_R = PA_i = 144{,}315 \text{ lb}$$

From this it is clear that pressure alone provides enough tension to overcome nearly half the required static compressive push force (381,840 pounds). However, because sinusoidal buckling can occur at net compressive loads of 17,000 to 35,000 pounds (see Section 5.4.2), pressure alone does not relieve enough of the compressive force to prevent buckling to drilling lengths of more than about 11,000 to 12,000 feet. Both continuous rotation and internal pressure are required to reach the desired long lengths.

5.3.1.1 Continuous Rotation Effects. Laboratory testing has shown that the coefficient of friction drops from the "static" value of 0.48 to a minimum dynamic value of about 0.33 as the rate of rotation increases up to about 5 rpm. This generates a tangential velocity of the pipe outer surface, transverse to the pipe axis, of 74.6 in./min. The coefficient of friction doesn't increase significantly until rotation increases to above about 10 rpm. A value of 5 rpm has been chosen as the nominal operating condition, with increases up to 9.2 rpm used during steering operations to provide commands to the steerable drillhead. The rotational effects in the following analysis only get better as rotation rate is increased. Thus, 5 rpm is chosen as a baseline value.

Figure 14 shows the vectors for the velocity of a typical element on the bearing/bottom surface of the pipe. In the nonrotating case, V_R is zero and all the velocity is along the pipe length. If the pipe could be pushed forward at a speed of about 75 in./min, the coefficient of friction would likely drop to about 0.33. However, available power limits the maximum drilling advance speeds to 24 to 36 in./min, with 6 to 12 in./min being much more common. Without rotation, the coefficient of friction remains high and the push forces large.

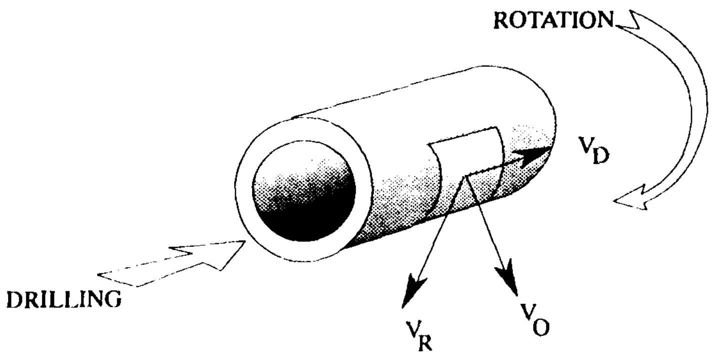
In the nondrilling case (during logging, while adding a new pipe section, etc.), the velocity vector is entirely rotational (V_D = zero).

The key assumption in the analysis is that the net forces on the bearing surface are the same <u>regardless of the direction of movement</u>. That is, the magnitude of the vector sum of F_D and F_r is a constant:

$$|F_0| = |F_D + F_r| = \mu\omega_0 L$$

Next, the vector addition assumption implies that:

$$F_D = F_0 \sin\alpha \quad \text{and} \quad F_r = F_0 \cos\alpha$$



@ 5rpm, & O.D. 4.75 in, V_R = 74.6 in / min

$V_O = V_D + \bar{V}_R$

Figure 14
Surface motion vectors.

where, since the forces must be along the line of the velocities,

$$\alpha = \tan^{-1}(V_D/V_R)$$

Table 8 shows the relationships between rotation rates, drilling speeds, and resultant push forces and torques. Figure 15 plots the relationships. For a typical rotation rate of 5 rpm, and a V_R of 74.6 in./min, drilling speeds (V_D) of as much as 36 in./min imply that:

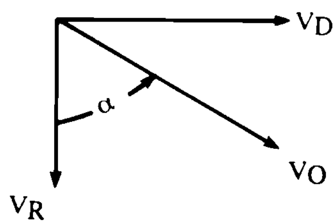
$$\alpha < 26 \text{ degrees}$$

so that the resultant push force is only about 43 percent of what it would be without rotation (10.5 lb/ft), even assuming the coefficient of friction remained at 0.33. At a normal drilling speed of about 12 in./min, that percentage drops to 16 percent, and at hard-rock speeds of 3 in./min the percentage drops to only 4 percent. Push forces can become very low at small advance speeds. In fact, the net push force required for 25,000 feet of pipe at even the maximum speed of 36 in./min is less than the rabbit force (144,000 pounds) associated with 15,000-psi internal pressure.

AS LONG AS THE PIPE IS ROTATING CONTINUOUSLY AT ABOUT 5 RPM AND UNDER FULL INTERNAL PRESSURE, FRICTION-INDUCED BUCKLING IS NOT A PROBLEM FOR NORMAL DRILLING SPEEDS OUT TO 25,000 FEET.

Table 8
Angle Between Rotational Velocity and Forward Velocity

Forward Velocity (V_D) (in./min)	α (deg) at 1 rpm $V_R = 14.9$ in./min	α (deg) at 2 rpm $V_R = 29.8$ in./min	α (deg) at 3 rpm $V_R = 44.8$ in./min	α (deg) at 4 rpm $V_R = 59.7$ in./min	α (deg) at 5 rpm $V_R = 74.6$ in./min	α (deg) at 6 rpm $V_R = 89.5$ in./min
2	7.63	3.83	2.56	1.92	1.54	1.28
4	15.01	7.63	5.11	3.83	3.07	2.56
6	21.90	11.37	7.63	5.74	4.60	3.83
8	28.20	15.01	10.13	7.63	6.12	5.11
10	33.83	18.52	12.59	9.51	7.63	6.37
12	38.80	21.90	15.01	11.37	9.14	7.63
14	43.17	25.13	17.37	13.20	10.63	8.89
16	47.00	28.20	19.67	15.01	12.10	10.13
18	50.34	31.09	21.90	16.78	13.56	11.37
20	53.27	33.83	24.07	18.52	15.01	12.59
22	55.85	36.40	26.17	20.23	16.43	13.80
24	58.13	38.80	28.20	21.90	17.83	15.01
26	60.15	41.06	30.15	23.54	19.21	16.19
28	61.94	43.17	32.02	25.13	20.57	17.37
30	63.55	45.15	33.83	26.68	21.90	18.52
32	65.00	47.00	35.56	28.20	23.21	19.67
34	66.30	48.72	37.22	29.67	24.50	20.79
36	67.49	50.34	38.80	31.09	25.76	21.90
38	68.56	51.85	40.33	32.48	26.99	23.00
40	69.54	53.27	41.78	33.83	28.20	24.07



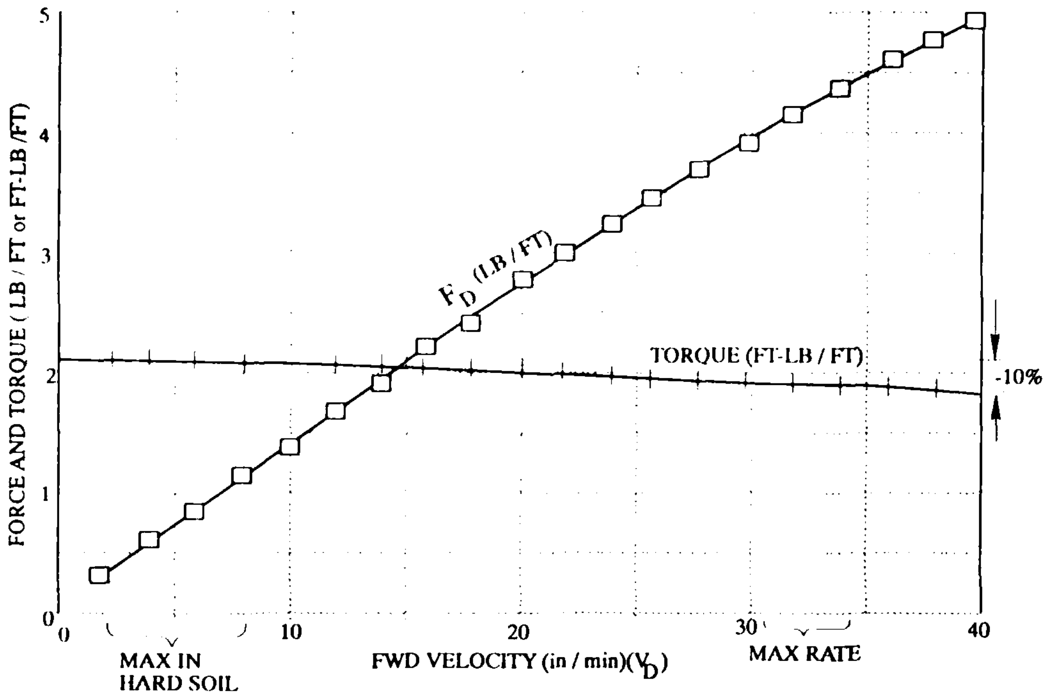


Figure 15
Push force and torque versus FWD speed.

Of course, when pressure must be kept low to minimize joint stresses, or when there are other loads such as those from collapsed holes or climbing pipes, or steering events, buckling can still be a problem. These effects are discussed in the following sections.

5.3.1.2 Internal Pressurization Effects on Stiffness. The Appendix provides the details of an analysis of the geometric stiffening effect from internal pressure in the pipe. The net effect of internal pressure is increasing the apparent stiffness (EI) of the pipe. Thus, greater bending force is required to produce a given steering effect in the pipe. That in turn means more resultant friction to resist rotation and forward movement. The effect of internal pressure is to make the pipe behave more like a tensioned cable than a simply-supported beam. The resultant pipe deflection for a given set of applied loads is one to two orders of magnitude smaller when the pipe is tensioned (by internal pressure) than when it is not. The pressure effect is clearly the dominant effect in controlling pipe steering and estimating loads from steering events. Of course, these effects vanish when the pressure is reduced for adding pipe sections, logging, etc., but the stiffness increases to its former value as soon as the pipe is pressurized again.

The geometric stiffness factor (ϕ) appears as a multiplier on the basic beam equation for the angular rotation per unit length at the end of a uniformly loaded beam (Ref 8):

$$\theta/L = \{\omega L^2/24EI\}\,[\phi]$$

where $\phi = 3(u\text{-}\tanh u)/u^3$, and $u = (PA_iL^2/4EI)^{1/2}$ with the other variables as defined in Section 5.1. Note that this factor is a function of length and axial force (PA_i). For pipe less than about 10 feet long, the pressure stiffening effect is negligible. However, for lengths of about 200 feet, the stiffness effect from axial tension is about two orders of magnitude (x 100) larger. Table 9 is a listing of the variables u and ϕ as a function of length. These corrections are used for all steering effect calculations.

Table 9
Geometric Stiffness Factors (u) and (ϕ) Versus Length (ft)
($u = (PA_iL^2/4EI)^{1/2}$, $\phi = 3(u - \tanh u)/u^3$)

For P = 15,000 psi, on the baseline pipe design, the following values are derived:

L (ft)	u	ϕ
10	0.991	0.7189
20	1.982	0.3927
30	2.974	0.2257
40	3.965	0.1427
50	4.956	0.0975
60	5.947	0.0706
70	6.938	0.0533
80	7.930	0.0417
90	8.921	0.0335
100	9.912	0.0275
125	12.390	0.0180
150	14.868	0.0127
175	17.346	0.0094
200	19.824	0.0072

5.3.2 Upslope/Downslope Drilling. Figure 16 shows the gravity and resultant friction forces on a straight pipe in an inclined hole. The F_g force is unaffected by rotation and internal pressure. However, the F_D force is affected by rotation as described in Section 5.3.1.1. Table 10 lists the upslope/downslope component of gravity as a function of slope angle. For comparison, it also lists the required F_D for a normal drilling speed (12 in./min), first at no rotation and then at 5 rpm. It can be seen that the upslope/downslope component of gravity can be a considerable fraction of the total loading on a sloped section.

This analysis may be used and the results added to the effects of curved sections in the vertical plane by calculating the slope (γ) and length (L) using only the end points of the curved section. The net effect of the gravity component is path independent. That is, the force derived by treating the pipe as a straight line from start to finish of the curve is the same as the value determined by integrating the effects along the curve.

The typical net slope required for most operational sites (end-to-end) is -0.5 to -5.0 degrees. That means that the net effect of the geometry is favorable, providing some natural additional distributed load on the pipe to move it forward without launcher force. Most of the maneuvers required will be well within ±10 degrees of the horizontal.

Table 10
Upslope/Downslope Gravity Loads on Pipe

Slope (γ) (deg)	Up/Downslope (F_g) (lb/ft)	F_D at 0 rpm (lb/ft)	Percent Change in (F_g + F_D)	F_D at 5 rpm (lb/ft)	Percent Change in (F_g + F_D)
+10	5.54	10.34	54	1.64	337
+5	2.77	10.47	27	1.66	167
+2	1.11	10.50	11	1.67	66
+1	0.55	10.50	5	1.67	33
0	0	10.50	0	1.67	0
-1	-0.55	10.50	5	1.67	33
-2	-1.11	10.50	11	1.67	66
-5	-2.77	10.47	27	1.66	167
-10	-5.54	10.34	54	1.64	337

In summary, since the pipe is heavy, the coefficient of friction is relatively low, and rotation keeps required push forces even lower, the slope effects of gravity can be significant portions of the pipe loading in planning/controlling push forces and joint loading.

5.3.3 Steering Effects. The most vital s eering actions for HDS technology are in the vertical plane. The drill string must be controlled to much tighter tolerances in the vertical plane than in the horizontal. This is particularly true in crossing under critical features such as buried facilities or a beach, and when making the final exit onto the seafloor. A depth error of 5 to 10 feet can make a big difference in the surface effects of the drilling process, weight of collapsed soil, etc. Such errors can also produce a linear distance error of many hundreds of feet on most gently sloping seafloors. By contrast, if a hole arrives within 500 to 1,000 feet of an offshore target horizontally, it will likely be well within specifications. Even when there are multiple installations planned at a candidate site, the horizontal circular target of ±1,320 feet should be adequate.

Fortunately, gravity aids in the vertical bending of the drill string, so vertical steering can be accomplished over reasonable ranges of angles with virtually no additional net frictional force to increase torque or push forces. The local loading will be increased, any effects from change of slope must be added, and there will be some increase in local bending stresses - but the net torque and push forces will not increase. The following analysis shows how this is accomplished.

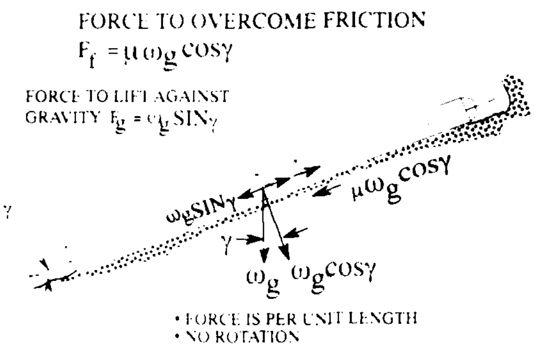


Figure 16
Forces on straight pipe, inclined hole.

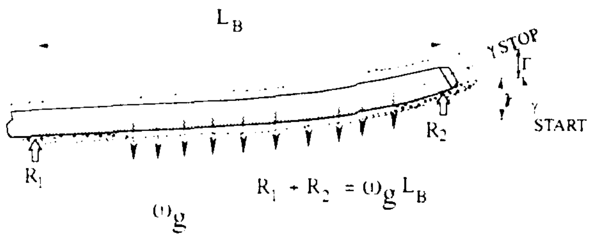


Figure 17
Forces on pipe, vertical steering upward.

5.3.3.1 Upward Steering. Figure 17 shows a sketch of a drill string being steered to climb, or increase the slope of the hole. The steering is being performed in such a way that the steerable drillhead is riding up on the cuttings at the end of the hole. The process follows a relatively large radius of curvature and the hole is large enough so the pipe is not forced up against the top of the hole. The result is that the pipe is suspended along all or part of its curved length. If the pipe rests on the cuttings along all of its length, the calculations for F_D and T are the same as for a straight pipe (adding any net slope effects). However, if we assume the pipe is supported only at its two ends, the question then becomes: What steering rate is allowable before the pipe is being pressed against the top of the hole and additional frictional forces are being generated?

In the analysis, it is assumed that the hole is essentially the same diameter as the pipe. As long as the steering produces a hole profile that has a curvature no greater than the natural curvature assumed by the simply-supported pipe, the pipe will follow the curve under the influence of gravity alone and will not press against the top of the hole. The following calculations therefore compute the predicted shape of the simply-supported pipe as a function of length. The results are displayed in terms of total vertical offset of the steerable drillhead from the initial pipe axis as a function of distance, assuming the pipe is horizontal to start. The value

z_o is for no internal pressure and z_p is for full pressure. The calculations also determine the angular change in slope (Γ) that is achieved by this steering action. The results are tabulated in Table 11.

The equation relating pipe bending angle to length and pipe characteristics is developed in the Appendix. The equation is:

$$\theta = \{\omega L^3/24EI\} [\phi]$$

As previously discussed, calculations assume full internal pressure in the pipe (15,000 psi), so:

$$u = (8.26 \times 10^{-3})L$$

For small angles, $\theta = z/L$ and $z = \theta L$. Thus,

$$z_p = \omega_g L^4/24EI(\phi) = (3.61 \times 10^{-7}) L^4 (\phi)$$

Table 11
Upward Vertical Steering Limits

Parameter	L (ft)				
	30	60	90	120	150
z_o (ft)	0.29	4.68	23.7	74.9	*
z_p (ft)	0.07	0.33	0.8	1.5	2.3
Γ (deg)	0.25	0.63	1.0	1.4	1.8

*Pipe will fail under these conditions.

One way to determine the required change in slope is to consider various possible geometries of launcher position, elevation, and distance from the beach. It is clear that in any operation there must at least be a change from negative slope to nearly level as the string crosses under the beach. Figure 18 and Table 1 outline some possibilities. For purposes of analysis, the figure assumes the string must pass at least 40 feet below the beach level. The figure also includes an estimate of the required distance to achieve the change in slope, derived from the values in Table 11 above. For assumed initial starting slopes of -2, -5 and -7 degrees, Table 1 shows the required standoff distance (Y) as a function of the elevation of the launcher above the beach (Z).

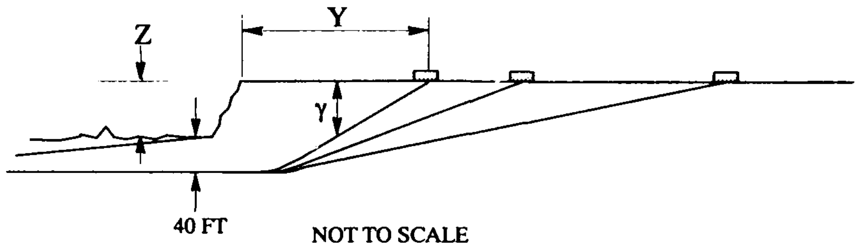


Figure 18
Examples of required launcher setback from beach.

Note that for the higher elevations there are some significant advantages to starting the hole with a 5- to 7-degree downslope. The standoff distance (and resultant hole length and cost) is decreased by over 4,000 feet. However, for small elevation changes, the table shows that a 7-degree downslope actually requires a longer standoff than 5 degrees (because of the added length required to correct the 7-degree downslope). The conclusion is that a net course correction of 5 degrees will probably be adequate for most applications.

In summary, the addition of pressure significantly reduces the amount of vertical steering that can be achieved without pressing the pipe against the top of the hole and adding friction forces. However, slope changes of from 0.5 to 2.0 degrees will probably be the largest that are desired in a single steering event. Experience in testing has shown that it is relatively easy to oversteer. The steerable drillhead cuts a hole that is sufficiently off round to force the drill string to bind up if it is fed too fast. Even the "natural" climbing tendency of the drill string produced slope changes of 1.4 to 5.7 degrees in about 400 feet. Therefore, although the gravity sag of the pipe allows ample slope changes without additional frictional forces, any vertical steering beyond those small limits will quickly produce large increases in frictional loads and should be avoided.

5.3.3.2 Downward Steering. The downward steering analysis is very similar to the upward except that the drill pipe is treated as a cantilevered beam instead of a simply-supported one. The idea in this case is to allow gravity to force the steerable drillhead end down without the need to force the end of the drill string against the top of the hole. Figure 19 shows the assumed configuration. In the cantilevered case, the baseline equation for deflection at the end is:

$$z = \{\omega_g L^4/8EI\}\ (\phi)$$

Note that this is three times the equivalent deflection for the simply supported case (eight in the denominator versus 24). Table 12 summarizes the deflection as a function of length for pressurized and nonpressurized cases.

The angle change from the start is:

$$\Gamma = z/L = \{\omega_g L^3/8EI\}\ (\phi)$$

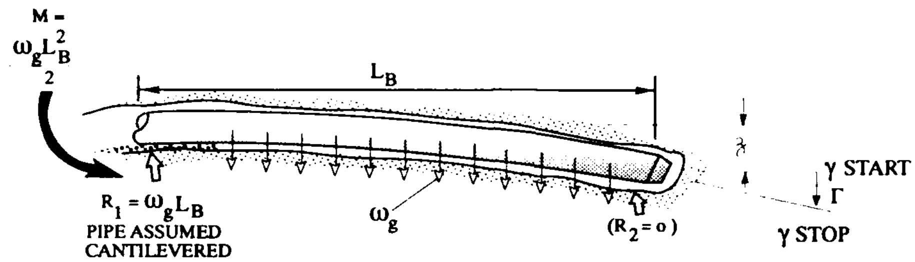


Figure 19
Forces on pipe, vertical steering downward.

Table 12
Downward Vertical Steering Limits

Parameter	L (ft)				
	30	60	90	120	150
z_o (ft)	0.88	14.00	71.0	*	*
z_p (ft)	0.20	0.99	2.38	4.47	*
Γ (deg)	0.38	0.95	1.51	2.13	*

*Pipe will fail in tension from bending at support point.

Although gravity produces much quicker downward bending than upward, there is a serious risk of pipe failure in this mode. The cantilevered weight of the pipe produces a tensile stress that will yield the pipe at a length of 72.4 feet, even without internal pressure. With pressure and no torque (near the drillhead), the allowable length reduces to about 60 feet. Even though the cuttings will distribute this load somewhat, it is clear that these bending stresses could rapidly overload the pipe joints that are already heavily stressed near the launcher. Therefore, the conclusion is that it is generally much better to start out with initial downslopes as large as required and make upward corrections later. This minimizes the number of downward steering events required and puts them out near the drillhead end (which has lower stress levels) rather than near the launcher. Since the pipe has generally shown a tendency to climb, it will be particularly important to monitor that effect. A situation where rapid downward steering is required is risky.

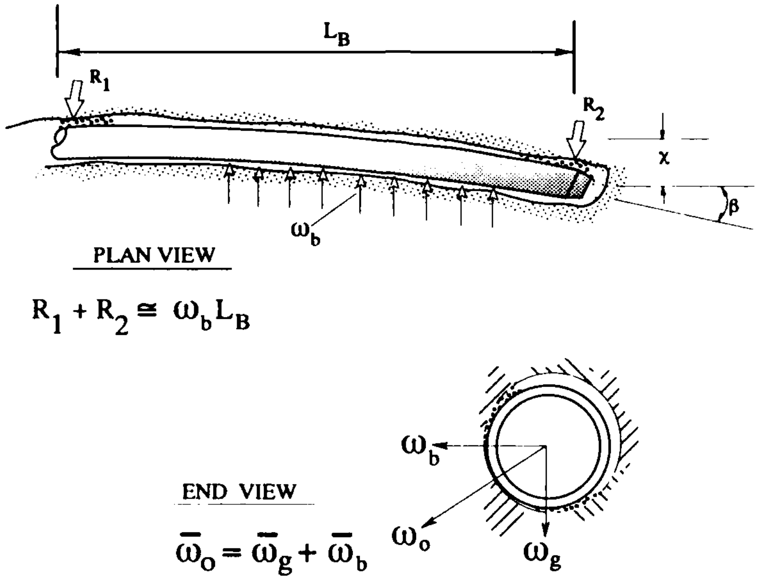


Figure 20
Forces on pipe, horizontal steering.

5.3.3.3 Horizontal Steering. Horizontal steering refers to corrections left or right of the baseline drilling track (cross-track corrections) in a horizontal plane. HORIZONTAL STEERING IS NOT POSSIBLE WITHOUT SOME ADDITIONAL FRICTION FORCES. The question is, how can those forces be minimized? The following analysis addresses that issue.

Figure 20 shows the assumed configuration. The pipe is curved in a horizontal plane. The pipe near the steerable drillhead is riding against the side of hole away from the turn. The pipe back from the end is pressed against the opposite side of the hole and even further back the pipe has a reaction similar to the drillhead end, on the same side of the hole. For purposes of analysis, the two end reaction forces are treated as point loads and the load along the pipe between them is considered uniformly distributed. In practice, neither will be strictly true. However, the integrated result of this - the frictional loads resisting pipe advance and rotation - is assumed to produce the same result regardless of the details of the loading. This will not be true if there is such a concentrated local load that more than simple friction occurs, but otherwise this assumption seems warranted.

Also, it is assumed that the pipe gravity loads are uniformly distributed; the pipe rests on or near the bottom of the hole along its length. Since the distributed bending loads ω_b are added vectorally to the gravity loads ω_g, the results are not sensitive to this assumption. The pipe could be pushed half way up the wall and the vector sums still apply.

The baseline equation is the same as the simply-supported analysis for upward steering. However, the analysis in this case assumes a steering event and solves for the distributed load that must be applied to cause that event:

$$\omega_b = 24EI\theta/\{L^3 (\phi)\}$$

Since $\beta = 2\theta$, substituting values for the pressurized pipe produces:

$$\omega_b = 768{,}761\beta/\{L^3 (\phi)\} \text{ lb/ft}$$

where β is in degrees and L is in feet.

Adding the bending forces and gravity forces vectorally,

$$\omega_o = (\omega_g{}^2 + \omega_b{}^2)^{1/2}$$

Then from statics, the sum of the reaction forces at the ends of the bend section (R1 + R2) must equal the product of the bending load and the length of bent pipe ($\omega_b L_b$). Thus, the total torque required to rotate the section in the bend is:

$$T_b = \omega_o L_b \mu r \text{ (ft-lb)}$$

Similarly, the push force (F_D) required to advance the pipe is increased by the bending load friction. As before, the push force is reduced by the continuous rotation of the pipe:

$$F_D = \omega_o L_b \mu \sin(\tan^{-1} V_D/V_R) \text{ (lb)}$$

Table 13 lists the values of ω_b, ω_o, T_b, and F_b for various horizontal steering angles (β) and distances (L_b). The results are plotted in Figure 21. The table and plot show that the bending loads can easily become much larger than the gravity loads. Only in the case of very small steering angles (less than 1 degree) over lengths of more than two or three pipe sections do the bending loads become less than the gravity loads. For steering of 5 degrees in a single pipe section length, the bending forces become 20 times as large as the gravity forces.

Fortunately, Figure 21 does show that there is a "best length" for any given desired steering angle. The general guidance is that it is best to use about 25 to 30 feet of pipe for every degree of steering - 1 degree per pipe section. Any length shorter than that produces high bending loads. Any length longer than [illegible] simply adds more frictional loads because of the longer pipe. As long as the loads are distributed over about 30 feet per degree, the net effect will be kept as small as practical.

Some other observations from these data are that it requires about 250 ft-lb of torque per degree of steering, even if the steering is carefully done. This represents about 1 percent of the total torque capacity of the joint, or 1 percent of the allowable total length of the drill string.

IT IS THEREFORE HIGHLY DESIRABLE TO LIMIT HORIZONTAL STEERING TO A MINIMUM. It also would follow that it is best to make horizontal steering corrections early so they can have the maximum effect on final positioning. The system also can be most effective if the offshore targeting is selected to allow maximum freedom in cross-track positioning, even if that adds some restrictions on the vertical or along-track target envelope. These ideas also influence the overall logging scenario. The accuracy of the logging system in measuring cross-track position degrades linearly with distance, which further suggests that cross-track horizontal steering be minimized. It is not practical to correct that which cannot be accurately measured. By contrast, the vertical position can be measured by pressure differential, which is not degraded by distance. In fact, if there were a pressure sensor in the drillhead that could telemeter data

during section change outs (by acoustic pulses, for example), it might well be possible to drill the entire hole with only one or two other logging runs. With experience, it might be possible to eliminate logging altogether except for the depth readings.

Table 13
Horizontal Steering Loads

Parameter		Horizontal Steering Angle (β) (deg)			
		0.5	1.0	2.0	5.0
L = 30 ft:	ω_b (lb/ft)	63	126	252	631
	ω_o (lb/ft)	71	130	254	632
	T_b (ft-lb)	132	241	472	1173
(@36 in./min)	F_D (lb)	306	559	1093	2713
L = 60 ft:	ω_b (lb/ft)	25	50	101	252
	ω_o (lb/ft)	41	60	106	254
	T_b (ft-lb)	152	223	394	943
	F_D (lb)	353	516	912	2186
L = 90 ft:	ω_b (lb/ft)	16	31	63	157
	ω_o (lb/ft)	36	45	71	160
	T_b (ft-lb)	201	251	395	891
	F_D (lb)	465	581	916	2065
L = 125 ft:	ω_b (lb/ft)	11	22	44	109
	ω_o (lb/ft)	34	39	54	113
	T_b (ft-lb)	263	302	418	874
	F_D (lb)	610	699	968	2026

5.3.4 Collapsed Hole Effects. It is much more difficult to predict the loading in a collapsed hole than in an open hole. The reason is that the loading in a collapsed hole depends on: (1) the density of the soil, (2) the effective depth of the collapsed soil (the weight of the column of soil acting on the hole), and (3) the length along which the hole is collapsed. Even though the coefficient of friction is essentially the same as for an open hole, the range of possible variation in density, effective depth, and collapse length produces a wide range of possible cumulative loading on a pipe from a collapsed hole event.

The baseline equation for loading is:

$$F_{fmax} = \mu d \rho_s D_o \text{ (lb/ft)}$$

Typical values of ρ_s range from about 30 lb/ft^3 for wet soil to 144 lb/ft^3 for crushed rock. It is likely that when a hole collapses into such a small volume as the space around the pipe, only a few feet of soil or rock above the pipe will be affected, or possibly even only a few inches. However, when drilling in a soft material such as soft clay with low cohesive strength (as when

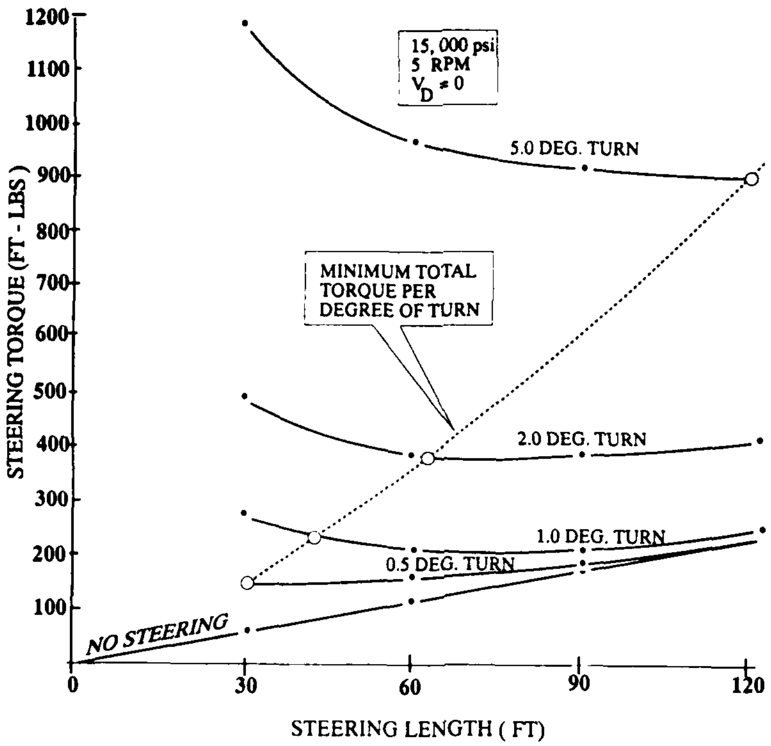


Figure 21
Horizontal steering torque versus length and angle of turn.

nearing the surface), there easily could be a collapse that affects soil all the way to the surface. Therefore, Table 14 addresses the per foot loading that will occur if soft material (30 lb/ft^3) collapses all the way to the surface as pipe is drilled beneath the seafloor for distances up to 80 feet.

Using $\mu = 0.33$ and a maximum speed of advance of 36 in./min, the following equations apply:

$$F_f = 3.92d \sin(\tan^{-1} V_D/V_R) = 1.7d \text{ (lb/ft)}$$

$$T_c = F_{fmax} r = 0.776d \text{ (ft-lb/ft)}$$

Table 14
Friction Loading in a Collapsed Hole
(ρ_s = 30 lb/ft^3, V_D = 36 in./min at 5 rpm)

Parameter	Effective Depth of Collapse (d) (ft)							
	10	20	30	40	50	60	70	80
Torque (T) (ft-lb/ft)	8	16	23	31	39	47	54	62
Push Force (F_f) (lb/ft)	17	34	51	68	85	102	119	136

It is clear that if the depth of collapse is more than a few feet, or if the length of the collapse is more than a few feet, the increment of torque and push force can become very large. For comparison, the torque induced by straight and level drilling is about 2.1 ft-lb per foot and the torque induced by a 1-degree horizontal steering event in 30 feet is about 8 ft-lb per foot. From Table 14 above, even a collapse of about 10 feet of soil produces a per foot torque loading equal to the steering event. Collapse of soil along even a few feet of the hole becomes significant if the system is attempting to reach long distances.

In general, there is no independent way to determine the extent of a collapsed hole, or to do much about it after the fact. The net effect is an increase in torque and push force with very little change in drilling length. The difference between this effect and an oversteering load is that if the drill string is pulled back, an oversteering load goes away; the collapsed hole problem does not. Both a too-curved hole and a collapsed hole may respond to being washed out and corrected. The difference is that a straightened hole will stay straight, whereas a collapsed hole may just continue collapsing. The only remedy for a collapsed hole is casing (if close to the launcher) or use of drill mud to seal off and support the collapse (or prevent it in the first place). If these methods aren't effective, the only choices are to exit early, or drill another hole elsewhere.

5.4 Analysis of Pipe/Joint Reactions to Loads

There are two likely failure modes for the drill string. They are both caused by the stresses induced from friction with the soil or rock and water pressure inside the pipe with the friction-induced stresses dominating. At present, the most likely limiting factor is the stress on the joint. References 4, 5, and 6 provide detailed analyses of the joint development, analysis, and testing.

All of the testing and all of the analyses agree that the present joint design is capable of functioning under the combined load of about 20,000 ft-lbs of torque at 15,000-psi internal pressure. This is a factor of more than two better than any commercially available joint design; it will allow drilling operations out to 10,000 feet. The most recent tests (Ref 9) showed the joints would operate up to 37,000 ft-lb of torque at zero pressure with minimal yielding.

The following discussions summarize the behaviors and failure modes of the joint as they are presently theorized.

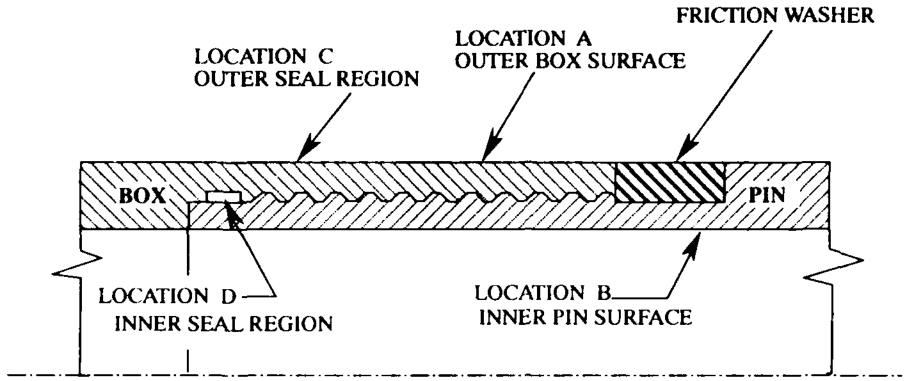


Figure 22
Joint design with possible failure regions.

5.4.1 Pipe Joint Limitations. Figure 22 shows a cross section of the joint design. Loads are transmitted through the joint along two paths:

a. Through the pin and threads to the box.
b. Through the friction washer directly to the box.

Since it is desirable to keep the joint stresses below yield, the material is assumed to always perform elastically so the effects of different loading forces can be analyzed independently and added linearly. The primary forces acting on the joint are:

a. Internal pipe pressure (P)
b. Torque (T)
c. Drilling push or pull forces (F_D)
d. Bending forces (ω_g, ω_b, and reaction forces)

The internal pressure produces axial tensile stresses that are essentially the same as those that would be produced by a tensile load from the launcher. Pressure also produces hoop stresses (tension). Note that the axial tensile stresses are carried through the pin via the threads to the box. No tension is carried through the friction washer, by definition. The internal pressure-induced hoop stresses are carried by the pin and the box. No hoop stresses are carried in the friction washer. The analysis of these effects is reasonably straightforward. These stresses are generally relatively small compared to the stresses generated by the applied torque.

Applied torque produces shear stresses that are easily predicted in the main pipe section. However, prediction of the shear stresses carried through the friction washer, the unthreaded section of the pin, and through the various threads to the final loaded condition on the box section is very difficult. It depends on the coefficient of friction between the friction washer and

the pin/box shoulders and the coefficient of friction on the threads. In general, it is very desirable to have a high coefficient of friction on the friction washer (hence its name), and a low to moderate coefficient of friction on the threads.

The primary problem is that if the coefficient of friction of the washer is low, a much larger compressive force must be applied to the box section to transfer the same applied torque through the joint than that required for a washer with a large coefficient of friction. This in turn causes a large axial tensile force on the pin. The larger compressive stress together with the torque-induced shear stress will result in a larger principal stress in the box for a given applied torque. Thus, the box section will yield a much lower torque for the washer with the lower coefficient of friction.

The Western Instrument Corporation analysis focused on the pin section and concluded that adding compressive loads (through additional push force on the pipe) would help because it counteracts the tensile loads on the pin. However, any addition of compressive loads also increases the friction washer-induced compressive force on the box. Results from the NCEL analysis indicate that the box is likely to fail before the pin because the safety factors that must be applied for compressive loading to prevent localized buckling are greater than in tension. The tests performed to date seem to support that conclusion because the joints generally yielded first by bulging (compression) near the box shoulder rather than in tension.

The problem of analyzing the joints analytically or experimentally is complicated by the large variability of the apparent coefficient of friction. The coefficient of friction seems to vary with the design of the washers, as might be expected. However, it also seems to vary from test to test and with loading as the washer seats and microyielding of the surfaces occurs. It was speculated that the large variability is caused by the uncontrolled makeup torque on most of the joint specimen tests. Using a makeup torque about one-half of the yield torque capacity of the box should significantly reduce this variability (see Ref 9). Unfortunately, most of the initial tests did not directly instrument the shear strain on the joints as a function of loading. Thus, attempts to infer coefficient of friction from the available data were not conclusive. Estimated coefficients of friction varied from a low of 0.1 to a high of 0.6 for the friction washer. This is not sufficient because the joint performance is essentially linear with this number. For example, if the coefficient of friction were 1.0 (the pin and box were welded together with a full penetration weld), the joint capacity is between 60,000 and 70,000 ft-lb of torque at full pressure (depending on what safety factor is selected). This is 20 to 25 percent more than is likely to be needed to reach a distance of 25,000 feet. However, if the coefficient of friction is in the more probable range of about 0.4, the torque limit would be about 20,000 ft-lb, for a distance of 10,000 feet.

Recently, three pipe joint specimens were monitored with 45-degree strain rosettes and tested under internal pressure only, torque only, and torqued to failure without internal pressure (Ref 9). Analysis of the test results by NCEL indicates that the joint would start to yield when subjected to a pure torque of about 40,000 ft-lb. Under internal pressures of 10,000 psi and 15,000 psi, but without externally applied thrust, the joint would yield at applied torques of 38,000 ft-lb and 37,000 ft-lb, respectively. With an applied internal pressure of 15,000 psi and a thrust of 105,000 ft-lb, the joint would yield at an applied torque of 30,000 ft-lb. A summary of the maximum pipe joint drill distance limits is given in Table 15. The maximum drill distances are given for 15,000-psi internal pressure, 5-rpm rotation, various advance speeds, and maximum torques of 25,000, 30,000, and 36,000 ft-lb. The corresponding maximum thrusts on

5.4.2 Pipe Buckling Limitations. There are three types of buckling that are of concern to the drill string:

a. Free-column buckling of the pipe sections in the launcher or between the launcher and the hole entrance.

b. Sinusoidal buckling of the pipe inside the hole.

c. Helical buckling of the pipe inside the hole (follows sinusoidal buckling if loading increases).

In the following subsections, it is assumed that the pipe joints have a negligible effect on the buckling strength of the pipe.

5.4.2.1 Free Buckling. The buckling of the free pipe in or around the launcher is of particular concern because it would be catastrophic (no restraint) and probably a serious safety hazard (high pressure water leaks, etc.). However, the problem can be contained by the use of pipe supports that are sufficiently close together. The present design has supports every 6 to 10 feet and that is considered adequate to handle the maximum expected loads during the development stage. Each time an operation is planned, it is important to determine the maximum push force that is planned (or possible) and ensure that the pipe is adequately braced against buckling. This analysis is performed using standard column equations for pinned ends. The analysis shows that the slenderness ratio must be greater than 46.0 for the pipe to fail by buckling. That is:

$$(l/r)_{min} = (\pi^2 E/135{,}000)^{1/2} = 46.0$$

This implies that the braced length must be <5.7 feet to prevent the pipe from buckling failure. Since the launcher is controlled to limit push/torque loads to prevent yielding of the pipe material, keeping spacing of the braces less than 5.7 feet will prevent buckling of those sections. Of course, this analysis neglects residual stresses in the pipe, the side loading effects of gravity, which would cause buckling sooner than if the pipe was vertical. This requires the free suspension length to be reduced to about a maximum of:

$$l_{max} = 3 \text{ feet}$$

This analysis is particularly important for the sections of pipe between the launcher and the hole entrance. It is also important in the design of the casing system and in determining the procedures for hole entry. If there are delays or repeated entries, it is possible to create large holes just inside the entrance. These could lead to long unsupported lengths and premature buckling failure. It is important to keep the entrance hole diameter small enough that free-column buckling is precluded.

the pipe joint that would cause yield initiation are 182,000, 104,800, and 11,900 pounds, respectively. The first row on Table 15 gives the maximum drill distance before yield initiation by torque alone without any thrust. The maximum drill distance is about 17,000 feet with a speed of advance of about 5 in./min or less. However, the HDS can operate at a maximum advance speed of 36 in./min. The required thrust for that speed will limit distance to about 15,000 feet. Even at a yield torque limit of 25,000 in.-lb, the HDS can reach 12,000 feet.

Table 15
Maximum Pipe Joint Drill Distance Limits
for 15,000-psi Internal Pressure

Rotation (rpm)	Advance Speed, V_D (in./min)	Maximum Drill Distance (ft)		
		T_{max}[a] = 25,000 ft-lb	T_{max} = 30,000 ft-lb	T_{max} = 36,000 ft-lb
5	0 to 5	13,000	15,000	17,000
5	36	12,000	14,000	16,000

[a]T_{max} = Maximum torque before joint yields, including stress from thrust at the length and advance speed shown.

It is concluded that the current joint design is capable of providing a maximum drill distance of about 17,000 feet under the most favorable conditions. It is speculated that the actual maximum drill distance attainable in the field is probably between 10,000 and 15,000 feet depending on field conditions encountered. This is about two to three times larger than that obtainable by conventional horizontal drilling.

All the drill distances shown in Table 15 are based on a coefficient of friction of 0.33. If the coefficient of friction can be reduced to 0.22 by the addition of a lubricant in the cutting fluid, the maximum drill distance can be extended to between 15,000 and 22,500 feet. Alternatively, a pipe with the same inside diameter but with a 5.0-inch outside diameter can be used for the last one-third of the drill string. Understandably, a friction washer with a slightly larger outside diameter would be needed. It is estimated that this would increase the average unit weight of the pipe drill string by about 14 percent and the torque capacity of the pipe joint by about 50 percent. The resulting net increase in maximum attainable drill distance is about 30 percent, giving an estimated drill distance of between 13,000 and 20,000 feet.

5.4.2.2 Sinusoidal Buckling. Sinusoidal buckling is a condition in which the pipe buckles into a sine wave shape inside the hole. It is controlled by the diameter of the hole, the outside diameter of the pipe, pipe unit weight, and pipe bending stiffness. For reasonably small hole sizes, sinusoidal buckling does not yield the pipe because lateral displacement of the pipe is limited by the hole diameter. Once the axial force is released, the buckling stops. Oil field experience suggests that a continuously rotating pipe will tolerate much higher push forces than a nonrotating pipe before sinusoidal buckling. However, for this analysis, this beneficial effect is neglected to provide a more conservative estimate. The critical net force required to produce sinusoidal buckling is:

$$F_s = (2EI\omega_g/3(D_h - D_o))^{1/2}$$

Table 16 summarizes the allowable net compressive push force for various hole diameters. Note that the values shown are for the pipe without internal pressure. Table 16 also lists the approximate length of pipe that can be pushed under various rotation speeds before the critical buckling load would be reached. If drilling speeds are kept slow (push forces reasonably low), it is possible to advance a rotating drill string at lengths of up to 25,000 feet in a nominal hole without sinusoidal buckling. However, without rotation, even a tight fitting drill hole will only support operations out to about 3,364 feet before buckling (which is of course very much like the present limit on commercial horizontal drilling).

Also, it appears that the degrading effect of increasing hole size is not as great as the improving effect of slower drilling speeds. The conclusion would be that it is better to advance slowly and risk a slightly larger drill hole than to advance faster and increase push forces.

Table 16
Sinusoidal Buckling Limits

Parameter	Hole Diameter (D_h) (in.)			
	5.5	6.0	7.0	8.0
Critical Push Force F_s (lb)	35,300	27,343	20,380	16,957
Critical Length (0 rpm) (ft)	3,364	2,606	1,942	1,616
Critical Length (6 in./min) at 5 rpm	41,932	32,480	24,209	20,143
Critical Length (36 in./min) at 5 rpm	7,734	5,991	4,466	3,716

The next step is to consider that with internal pressure added, the pipe is in 144,000 pounds of pretension. Unless there are a lot of horizontal steering events or collapsed holes, a full 25,000 feet of pipe can be pushed at 36 in./min (at 5 rpm) and only require about 114,000

pounds of push force. The net force on the pipe is therefore still in tension and the pipe cannot buckle. This does not include the beneficial geometric stiffening effects from the pressure.

This encouraging thought must be balanced against the observation that once the system loading does get close to the 144,000 pretension limit, things must be watched very carefully. First, any reduction in pressure is a linear reduction in the allowed push force. Second, if there are areas of reasonably large hole diameter (which is most likely near the hole entrance, where forces are largest), it only takes about 17,000 pounds beyond the tensioning force to cause buckling. This is why the acceptable combinations of pressure, torque, and push force are programmed into the launcher controller. It would be virtually impossible for an operator to dynamically monitor all these variables and stay in the acceptable combinations.

5.4.2.3 Helical Buckling. Helical buckling occurs at a load about 40 percent greater than sinusoidal buckling. The pipe forms a complete helix along the interior surface of the hole. Friction forces increase rapidly, and the pipe can move to failure from this condition. The pipe generally has yielded to some extent if helical buckling occurs (not true with sinusoidal buckling). Since sinusoidal buckling is used as the definition of "failure," helical buckling will not be encountered except if there is a major equipment malfunction. Even then, the pipe probably can be recovered, or even used if the hole can stop there.

5.5 Sample Problem

The following sample problem is provided to illustrate the ideas set forth in the analysis section. Figure 23 is a cross-section sketch of the drilling profile. The example represents a site that is reasonably generic. The launcher start point is 7,500 feet back from the beach area and is at an elevation of 250 feet above Mean Lower Low Water. The seafloor slopes uniformly to reach a depth of 90 feet at a distance of 25,000 feet from the launcher site (17,500 feet offshore). That is the desired exit point. However, the at-sea interface operation can still occur if the exit point is in at least 60 feet of water. That makes the minimum distance required about 19,000 feet. This analysis assumes an improved joint capable of 52,000 ft-lb maximum, somewhat larger than the current joint capacity of about 40,000 ft-lb.

5.5.1 Planning Ahead for Exit. To minimize the number of steering events, the plan is to head first for a point 120 feet below the beach, then drill level to the start of the climb up to the seafloor. That will mean the hole will be 30 to 60 feet below the seafloor at the time the final climb is started, depending on the actual distance drilled to that point. The seafloor is generally rock but there are occasional sand pockets a few feet deep. Therefore, the final upslope section will be at an angle of +5 degrees. If the actual rise winds up being 30 feet (120 feet to 90 feet), the 5-degree angle implies a distance of 343 feet for the climb. In addition, Table 11 from Section 5.3.3.1 shows that an upward steering event of 5 degrees will take about 450 feet. That steering process produces a rise of about 5 feet, which reduces the required straight upslope distance to 286 feet. Therefore, for planning purposes, the final upward vertical steering event will start about 725 feet from the desired exit point if the drill string makes the full 25,000-foot distance. If the exit is at the minimum distance (60-foot water depth), the final climb will start (630 + 450) = about 1,080 feet from the exit.

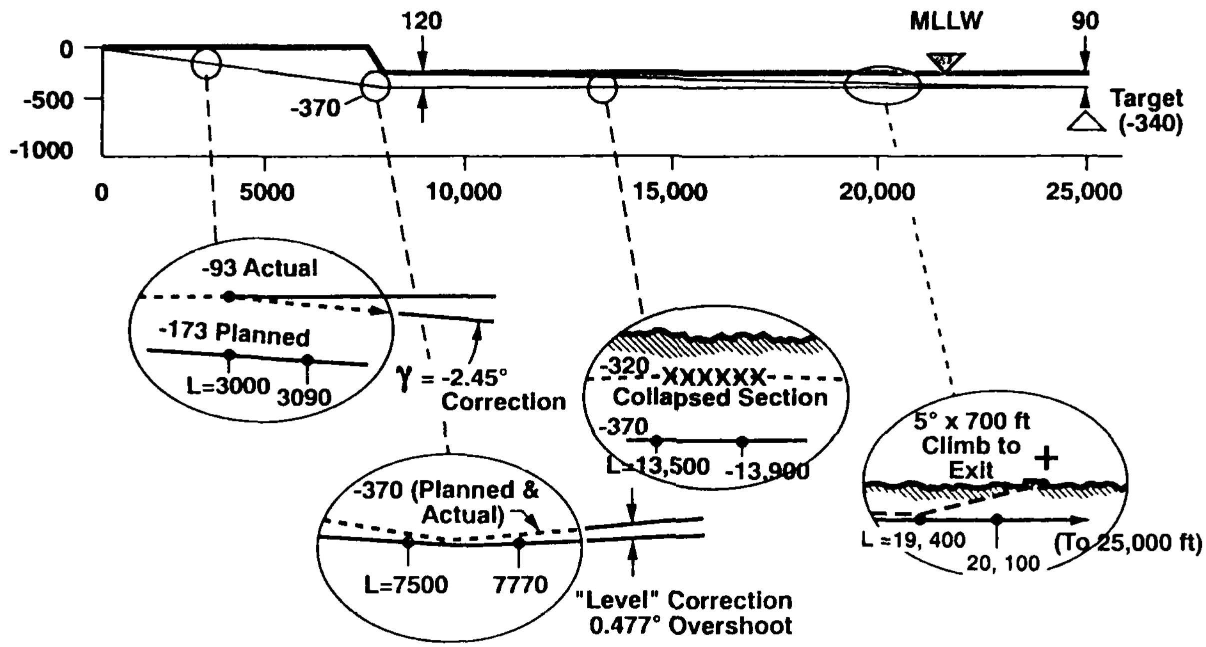


Figure 23
Sample problem No. 1.

Although the launcher control electronics are designed to protect the system from overloads, it is a good idea to look ahead in the drill plan and estimate the expected loads required to complete certain steps near the end of the operation. That way, if loads begin to increase during the early stages the operator will always be able to at least know how much additional torque or push force will be required to exit to the seafloor and complete the hole at whatever distance has been achieved. Alternatively, the operator will know in advance if loads have reached a level that make it impossible to reach the target distances at present speeds (push force levels). Decisions regarding rework of the hole, reducing speed, adding drill mud, or other processes can only be made if reasonable predictions have been made in advance. It is better to solve anticipated problems and analyze feasible options than to improvise on site while the system continues to rotate (flushing out a larger and larger hole) and the operating costs continue to mount.

Take, for example, the final climb. For the maximum likely length of 1,040 feet at 5 rpm, the required additional torque will be:

$$T_{climb} = \mu\omega_g Lr(\cos 5) = 2153 \text{ ft-lb}$$

That is about 4 percent of the total torque capacity of the joint; small, but enough to be monitored.

The required push force is due to a combination of the gravity effect of the slope and the frictional force - which is a function of drilling speed. For a speed of advance of 36 in./min (through soft soils), the push force could be as high as:

$$F_{climb} = [\omega_g(\sin 5) + \mu\omega_g \sin\{\tan^{-1}(V_D/V_R)\}(\cos 5)]L$$

$$= (2.77 + 4.55)1040 = 7609 \text{ lb}$$

Even at a very slow speed of advance of 3 in./min, the push force still needs to be at least 3,318 pounds, so there must be some reserve left when the decision to exit is made.

5.5.2 Leg One. The initial downslope steering angle to aim directly for the beach turning point is:

$$\gamma = \tan^{-1}(-370/7500) = -2.83 \text{ degrees}$$

Knowing the drill string tends to climb, the launcher is set up at a baseline downslope of -3.0 degrees. Drilling begins and continues to a distance of 3,500 feet. At that point, the hole is logged. The logging data indicate the hole is 150 feet left of and 80 feet above the desired line. The net downslope of the hole is -1.52 degrees instead of the desired -2.83 degrees. Assuming the hole entry has been cased and there are no collapsed holes, the likely loading on the pipe near the launcher at this stage of operations can be estimated using the same procedures applied to the final climb. That is:

$$T_1 = \mu\omega_g Lr(\cos\text{-}1.52) = 7271 \text{ ft-lb}$$

@V_{Dmax} (36 in/min): $$F_1 = [\omega_g(\sin\text{-}1.52) + \mu\omega_g \sin\{\tan^{-1}(V_D/V_R)\}(\cos\text{-}1.52)]L$$

$$= 18{,}922 \text{ lb}$$

5.5.3 First Steering Correction. The desired corrections to reach the beach turning point are:

Vertical:

Needed new	$\gamma = \tan^{-1}[(-370 - (-93))/4000] = -3.97$ deg
Present	$\gamma = -1.52$ deg
Correcting	$\Gamma = -2.45$ deg

Horizontal:

Needed new	$\beta = \tan^{-1}(150/4000) = 2.15$ deg
Present	$\beta = \tan^{-1}(150/3500) = 2.45$ deg
Correcting	$\beta = 4.6$ deg

From the charts on downward vertical steering (Table 12), a correction of -2.45 degrees could be achieved in 150 feet (five pipe sections) with no additional torque or push force. However, this would likely cause significant bending stress in the pipe, so the turn should be performed over at least seven pipe sections (210 feet).

Section 5.3.3.3 shows the horizontal steering could be performed in two 60-foot steps or one continuous 120-foot operation with about the same results. Present drillhead experience does not indicate if it is practical to change directions vertically and horizontally at the same time. Assuming it is not, the corrections may be performed in any order. It probably would be appropriate to perform both operations, using rotation-rate commands, to complete all the desired changes before attempting another logging run. In fact, since the hole at this point is well underground and clear of known obstacles, it may well be appropriate to make the steering changes and then drill "straight" on the new heading for a few hundred feet to clearly establish the new heading before spending time on another logging measurement.

In any event, the additional forces expected from these two steering events are as follows. For this analysis, the downward steering in 90 feet is first and the horizontal steering in 120 feet is second for a total of 210 feet of steering distance:

Vertical Steering - In vertical steering, no additional loads occur except basic friction loads, minus the average downslope effects. The starting slope is -1.52 degrees. The added drop required from steering down -2.45 degrees over 240 feet is approximately 4 feet, for an average added downslope angle of -0.95 degrees through the steering. The total average downslope during steering is therefore -2.75 degrees:

$$T_2 = \mu\omega_g Lr(\cos\text{-}2.75^o) = 498 \text{ ft-lb}$$

@V_{Dmax} (36 in./min): $$F_2 = [-\omega_g(\sin\text{-}2.75^o) + \mu\omega_g \sin\{\tan^{-1}(V_D/V_R)\}(\cos\text{-}2.75^o)]L$$

$$= (-1.53 + 4.56)240$$

$$= 727 \text{ lb}$$

Horizontal Steering - Although the horizontal steering occurs while the drill string is headed slightly down slope (-3.97 deg), the only effect is the nonfrictional gravity effect. The bending calculations are affected only by the cosine of the angle, which may be ignored because the effect is less than 0.2 percent, and the analysis is conservative. The tables in Section 5.3.3.3 show by linear interpolation that a horizontal steering event of 4.6 degrees in 120 feet produces the following:

$$\omega_b = 116 \text{ lb/ft}$$

$$\omega_o = 120 \text{ lb/ft}$$

$$T_3 = 876 \text{ ft-lb}$$

$$F_3 = 2032 \text{ lb}$$

Therefore, at the new length of 3,830 feet, the total estimated loads would be:

$$T_{3830} = 7271 + 498 + 876 = 8645 \text{ ft-lb}$$

$$F_{3830} = 18{,}922 + 727 + 2032 = 21{,}681 \text{ lb}$$

Of course, this is only an estimate. However, it is important to make such a running estimate and compare it with the measured values of the launcher control system. If the values are substantially higher than predicted, there may be a collapsed hole or other problem. If they are substantially lower (or higher), the instrumentation may be in error. In any case it is then necessary to check the instruments and plan for corrective action as required. For now, both of the numbers are well below any pipe failure limits, so drilling can proceed at whatever speed the formation will allow.

5.5.4 Leg Two. On the new vertical and horizontal headings, the drilling proceeds toward the target for the turn beneath the beach. To keep the problem simple, assume drilling stops when 7,500 feet of pipe is in the hole and the logging indicates the drillhead is precisely on target at the turn for the beach. The calculations for estimated loading on Leg Two are as follows. The slope is -3.97 degrees, the length is 3,670 feet:

$$T_4 = \mu\omega_g Lr(\cos\text{-}3.97) = 7611 \text{ ft-lb}$$

@V_{Dmax} (36 in./min):

$$F_4 = [\omega_g(\sin\text{-}3.97) + \mu\omega_g \sin\{\tan^{-1}(V_D/V_R)\}(\cos\text{-}3.97)]L$$

$$= (-2.2 + 4.55)3670$$

$$= 8625 \text{ lb}$$

The total loads at the turning point are now:

$$T_{7500} = 7271 + 498 + 876 + 7611 = 16{,}256 \text{ ft-lb}$$

$$F_{7500} = 18{,}922 + 727 + 2032 + 8625 = 30{,}306 \text{ lb}$$

5.5.5 Second Steering Correction. The desired corrections to reach level are:

Vertical:

Needed new	$\gamma = 0$ deg
Present	$\gamma = -3.97$ deg
Correcting	$\Gamma = +3.97$ deg

Horizontal: (none)

From Section 5.3.3.1, this turn can be made in a distance of about 360 feet (12 sections) without any additional bending frictional forces. The torque and push force loads may be simply added to the next leg as part of the length.

5.5.6 Leg Three. Assume that after another 6,000 feet the logging system shows the hole has climbed 50 feet and moved left 300 feet. The angular movement to the left is still well within the target. It would be nice to adjust horizontally toward the target but it would take about 5.63 degrees to aim directly at the target center. The present movement is off an average of 2.86 degrees to the left, and it is an additional 2.77 degrees to reach the target. This is a relatively large correction, so there needs to be a tradeoff between possible distance limits and desired horizontal accuracy. A lot would depend on the actual logging trace. If the hole has wandered off briefly but is not headed correctly, no problem. Also, if there is a general steady trend in this direction the string will likely hit the target. However, if there has been a strong change in heading that would lead outside the target there is no choice but to make a correction back to the right. For now, assume no horizontal correction is to be made.

The vertical movement, however, is another matter. The data mean the drill is within about 10 feet of the seafloor. That is probably too close, given the uncertainties in seafloor bathymetry data and the overall accuracy of the pressure sensor depth indication. A downward correction is required. The correction is:

Vertical:

Needed new	$\gamma = -0.27$ deg
Present	$\gamma = +0.48$ deg
Correcting	$\Gamma = -0.75$ deg

This is an easy correction. From Section 5.3.3.2 that correction can be made within a two-section length (60 feet) with little or no risk of additional frictional loads or significant pipe bending stresses. This amounts to only a few minutes of downward steering action by the steerable drillhead.

The loads that have been added by Leg Three are:

$$T_5 = \mu\omega_g rL = 12{,}468 \text{ ft-lb}$$

@V_{Dmax} (36 in./min):

$$F_5 = [\omega_g(\sin 0.48) + \mu\omega_g \sin\{\tan^{-1}(V_D/V_R)\}(\cos 0.48)]L$$

$$= (0.27 + 4.56)6000$$

$$= 28{,}980 \text{ lb}$$

At this point, before steering, the cumulative loads are:

$$T_{13,500} = 7271 + 498 + 876 + 7611 + 12{,}468 = 28{,}724 \text{ ft-lb}$$

$$F_{13,500} = 18{,}922 + 727 + 2032 + 8625 + 28{,}980 = 59{,}286 \text{ lb}$$

5.5.7 Collapsed Hole. To make the sample problem interesting, and possibly more realistic, assume that during the excursion near the seafloor and in making the downward steering turn, part of the hole collapsed. The actual symptoms observed would simply be an increase in torque and push force with little or no advance in the drill string. When the push force is relaxed, the torque still stays high. It is possible to estimate a little bit about what happened as follows:

GIVEN - Depth below seafloor (d) is about 10 ft
Wet soil density (ρ) is assumed 30 lb/ft^3
Observed increase in torque T_c is 4196 ft-lb
Observed increase in F_D at 36 in./min is 21,200 lb

THEN - From Section 5.3.4, $F_D = 1.7dL$, so $L = F_D/17 = 124$ ft
(the torque checks because $T_d = F_D r$)

This implies that about 124 feet of hole collapsed.

From this point on, the problem is to determine how much torque and push force budget is left. There is very little that can be done to reduce the torque loads, but the push force can be reduced by slowing down. The allowable torque on the joints is 52,000 ft-lb. We know we have to keep about 2,153 ft-lb of torque for the climb to the seafloor. We know the torque to rotate the pipe for the length drilled so far, plus the collapsed hole, is:

$$T_{13,500+} = 7271 + 498 + 876 + 7611 + 12{,}468 + 4196 = 32{,}920 \text{ ft-lb}$$

That leaves a balance of about 9,927 (52,000 - 39,920 - 21,530) ft-lb of torque. If there are no further collapses, that equates to about 4,777 feet of drilling before starting the exit climb.

That would imply a total distance of (7,500 + 6,000 + 4,777 + 1,040) = 19,317 feet, which just barely meets the required minimum distance to reach the minimum water depth.

To check on the push force, we can apply a similar logic. The present push force at maximum speed of advance is:

$$F_{13,500+} = 18{,}922 + 727 + 2032 + 8625 + 28{,}980 + 21{,}200 = 80{,}846 \text{ lb}$$

Adding the required force for the climb at full speed (7,609 pounds) to the estimated force required to push 4,777 feet at full speed (21,801 pounds) gives a total of 110,256 pounds. That is still less than the rabbit force at 15,000 psi, so there is no danger of buckling.

5.5.8 Sample Problem Conclusion. Although logging takes longer in the longer hole, it is more critical to have accurate data during the final stages of drilling. Therefore, the final stages of drilling should probably plan to have more frequent logging than the beginning stages. The most important information is the depth because the drill string is close to the seafloor. Horizontal information is of less value because the accuracy is degraded at the longer distance and by that time there is very little that can be done to correct horizontal position significantly. In this situation, a real-time vertical position sensor would be very valuable.

Assuming Leg Four goes nearly as planned, the final step is to check position just before the climb and adjust the intended steering accordingly. In some seafloors, it may be possible to simply keep going "straight" and let the bathymetry slope produce an exit naturally. In fact, that is the most desirable target. However, in the sample problem it is assumed that a climb is necessary.

The vertical steering action is as already analyzed. It produces no additional frictional forces. The climb is a straight run of a few hundred feet.

The breakout onto the seafloor may not be easy to detect from the launcher end. As the pipe pushes out onto the seafloor, frictional forces remain the same as when it is in the hole. The only real indicator is that it is possible to maintain forward speed with virtually no internal increase in push force and a low cutting pressure.

The caution in this situation is that at long ranges the launcher controller will not allow the system to apply high push forces without internal pressure. Therefore, when it is suspected that the pipe has exited, the best test is to release internal pressure and apply very small forward forces while continuing pipe rotation. In time, even at very slow speeds, it will be possible to continue moving the pipe forward without any cutting pressure.

In some cases, it may be possible to detect exit by a sudden increase in the drilling water flow for a given internal pipe pressure. This effect could be fairly pronounced if the final few feet are cut through competent rock. In that case there would be a lot of back pressure (low flow for a given internal pipe pressure). That would change when the back pressure is released (higher flow for a given applied internal pressure, up to the theoretical limits).

Even the test of forward motion with no internal pressure still doesn't absolutely guarantee the pipe is on the seafloor surface. In a layer of soft sand or silt it is possible to move forward slowly with only a few hundred psi internal pressure to "wash" the unconsolidated soil aside. However, it is the best that can be done from the launcher. The final verification of exit must be performed by divers at sea. They can confirm the exit by seeing the plume of mud and turbulence. They also must visually sight the pipe (while it is still rotating) in order to confirm that it is in an accessible location for the end preparation and attachment of the flexible pipe.

If there is a problem with the exit location conditions, there are several possible solutions. The drill string can be used to simply blast the way clear with a lot of flow (assuming the joints will allow the pressure with no forward movement). The drill string can be gently pushed forward to provide a longer exposed section for easier diver access. The drill pipe may even be pulled backward to move it back from some nearby obstacle. These decisions must be made while the string is still rotating and the launcher is functioning.

Only after divers confirm the proper location and satisfactory exit of the pipe can rotation be stopped. That completes the primary installation of the drill string. The next steps are the preparation of the sea end, attachment of the flex pipe, and installation of the cables.

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Appendix

GEOMETRIC STIFFENING EFFECTS OF INTERNAL PRESSURE IN DRILL PIPE

INTRODUCTION

Steering the drill pipe can add frictional loads because of the resistance of the pipe to bending and the resultant bearing loads (friction) against the walls of the drilled hole. That resistance and those loads are a function of the stiffness of the pipe. That stiffness is significantly increased if the pipe is internally pressurized. The correction factor to be applied to the normal beam equations is applied as shown in the following paragraphs.

EQUATION FOR DEFLECTION OF PIPE VERSUS APPLIED LOADS AND INTERNAL PRESSURE

From Reference 8 (in the main text of this report), the slope of the deflection curve at the end of a uniformly loaded tie rod (tension element) is as follows:

$$\theta = \{\omega_o L^3/24EI\}[3(u-\tanh u)/u^3] \qquad (1)$$

where $u = (SL^2/4EI)^{1/2}$, with S being the axial tensile force.

Note that the $\lim_{(u \to 0)}[3(u-\tanh u)/u^3] = 1$, and Equation 1 reduces to that for a uniformly loaded beam.

For a pressurized pipe,

$$u = (PA_iL^2/4EI)^{1/2}$$

For the HDS pipe, $A_i = 9.62$ in.^{2}, $E = 30 \times 10^6$ psi, and $I = 17.62$ in.^{4}. Therefore,

$$u = (4.55 \times 10^{-9}\ PL^2)^{1/2}$$

and for $P_{max} = 15{,}000$ psi,

$$u = 8.26 \times 10^{-3}\ L$$

where L is in inches.

Values of u and the expression $\phi = [3(u-\tanh u)/u^3]$ are shown in Table 9 in the main text of this report.

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